The
ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOL. X. No. 6 JUNE, 1949

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IN THIS ISSUE

IN introducing the new section PRODUCT DESIGN AND DEVELOPMENT it is our aim to promote thought conducive to good design by presenting the best practice in modern design in this country and comparing it with products of foreign origin. Design for appearance has acquired a new significance, but it should not be regarded as an end in itself. Certainly no improvement is to be gained by stressing the value of appearance, if, at the same time, functional requirements are not fully satisfied. To state this in different words, design procedure should not end at a technical solution dictated by functional considerations alone. Such considerations are indispensable and, indeed, still constitute the most essential part of the design process, but they are incomplete unless supplemented by aesthetic considerations; the more these two are organically united, the higher the overall merit of the product. Mr. T. Haas, A.M.I.Mech.E., A.F.R.Ae.S., has written a few notes for us on this subject, on page 209, and we are also fortunate in being able to present some American reflections on functional design by Professor Earl D. Hay of the Iowa State College.

In this Section we shall endeavour to describe and illustrate new designs conforming to the best industrial design practice both in this country and abroad. In the hope that our readers will find this Section to such advantage that they welcome its introduction, we, on our part, would invite co-operation from executives prepared to send us information on their latest products including case histories whenever available, with a view to publication in this journal.

* * *

The theory of automatic control systems is the subject of a series of articles by Dr. W. Oppelt, who has also written a book on this same subject. Material from two of Dr. Oppelt's articles, together with selected sections from his book, form the basis of our abstract entitled "Fundamentals of Automatic Control." This abstract, the first part of which will be found overleaf, presents a brief review of the theory of control elements, and gives details of the work done in Germany in this field since 1939. A description is given of the methods evolved for an analytical and synthetic study of open- and closed-loop systems. A graphical method for achieving multiplication and addition in a complex plane is indicated, and, furthermore, the abstract describes an apparatus with a transparent rotatable disc with logarithmic spirals which can also be used for this purpose. Dr. Oppelt has also enumerated a series of stability criteria based on graphical and analytical considerations which are, in certain cases, more readily applicable than the Hurwitz and Nyquist criteria. Examples of control elements of various types are given, and the article is concluded with a numerical example illustrating the application of the Cremer-Leonhard criteria.

* * *

The article by E. Niedermayer on rheological criteria for two-stroke cycle systems, the second part of which is given in this issue on page 193, provides further bases for the calculation of inlet and exhaust slots for a scavenge system designed to have optimum characteristics. The article comprises a numerical example for an engine with a scavenge blower, and is concluded by a description of methods by which the air supply ratio, the scavenge ratio, the effective air consumption and other factors can be determined experimentally. The required data can be obtained either directly on the running engine, or by investigating conditions in model scavenge systems. Six different models are described, which enable a study to be made of the scavenging process and the gas flow under stationary or cyclically variable conditions.

* * *

In recent years the mechanical properties of metals at low temperatures have attracted the attention of engineers and metallurgists. This interest in low temperature mechanical behaviour has been shown by investigators of commercial problems as well as by those concerned with fundamental research. The commercial interest in low temperature behaviour arises from the fact that many commercial uses for metals, such as liquid gas storage and high altitude flying, depend directly on the low temperature behaviour of metals.

In view of the great importance of research in this field, an apparatus for tensile testing of metals at sub-zero temperatures has been designed and built at Case Institute of Technology for use in conjunction with a special fixture to produce a maximum eccentricity of less than 0·002 inch in loading. A description of this apparatus will be found on page 199.

GERMANY

Fundamentals of Automatic Control

By W. OPPELT. (From "Fundamentals of automatic control," *Wolfenbütteler-Verlagsanstalt G.m.b.H.*, Hannover, 1947, 118 pp., 32 illustrations; "Graphical methods for complex multiplication," *A.E.Ü.*, Vol. 2, 1948, pp. 76-78, 4 illustrations; "New methods for assessing the stability of automatic control processes," *Die Technik*, Vol. 3, No. 7, July, 1948, pp. 312-314, 7 illustrations.)

1. INTRODUCTION

THE basic servomechanism is defined as a system of non-interacting orientated elements forming a closed loop, the chain being designed in such a way that static stability prevails*

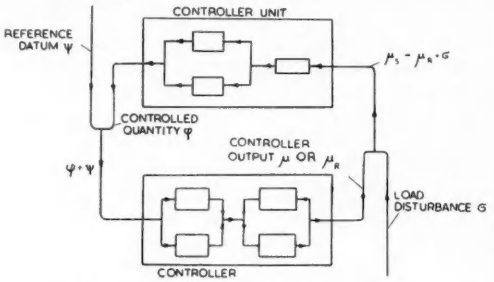

Fig. 1. **Structure of a simple control system.**

The block diagram of the control system is shown in Fig. 1. The structure is subdivided into a controller (the servomechanism proper), and a controlled unit (including the element subjected to automatic control).

2. NOTATION

φ = controlled quantity, ψ = reference datum, σ = load disturbance, μ_R = controller command, $\mu_S = \mu_R + \sigma$ = effective output or adjustment. The subscript R refers to the controller, S to the controlled unit, while e and a denote input and output variables.

In general, the quantities are expressed in non-dimensional form, and related to a given stationary condition, denoted by the subscript s. For instance, in the case of speed control, the controlled quantity is

$$\varphi = (n - n_s)/n_s = (n/n_s) - 1.$$

3. POSSIBILITIES OF CONTROL

The control system can be designed to operate on the basis of one of the following principles† :—

(1) measurement of the controlled quantity φ, and its derivatives with respect to time: φ', φ'', etc.; control direct from the controller R;

(2) measurement of φ, and of an auxiliary quantity φ_H taken from another point of the controlled unit S; control direct from R;

(3) measurement of φ, φ', φ'', etc.; control direct from R and also from an auxiliary control element with a command μ_H acting at another point of the controlled unit S;

(4) measurement of φ, φ', φ'', etc., and of the load disturbance (or disturbances) σ, at one or several points in S;

(5) measurement of φ from S, and of φ_M from a model reference system, the model having smaller time-constants and being subjected to the same load disturbances and commands as the controlled unit S (parallel networks).

Multiple-loop and cascade-control systems: In a multiple-loop system the controlled quantities are interacting, whereas in a cascade system the control arrangement is such that the first controlled quantity φ_I can affect the second network element, without it being possible for φ_{II} to act through a feedback on the first network element.

The complete control system comprises a measuring device or pick-up, differentiating or integrating elements, transmission elements, power amplifiers and relays, servomotors, reference-datum controllers, and feedback and smoothing networks. The pick-up is a measuring device sensitive to variations of a physical or chemical quantity, and it has a tapping connecting it to the input side of the controller R.

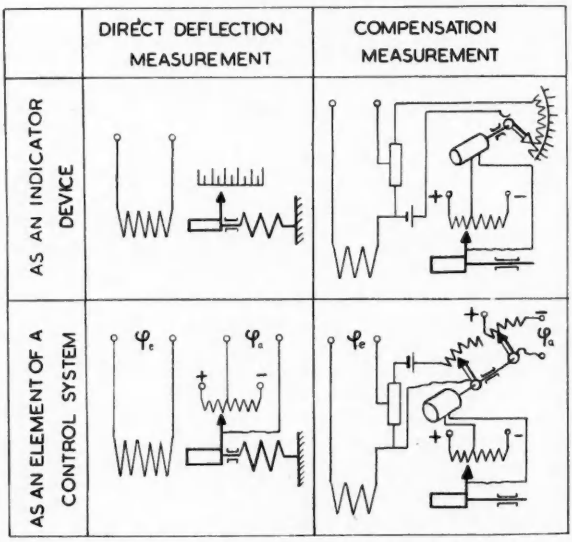

Fig. 2. **Indicators and pick-ups for measuring electrical quantities.**

The difference between a pick-up for automatic control and an indicating instrument is shown in Fig. 2. A number of differentiating elements giving derivatives of quantities with respect to time are illustrated in Fig. 3. Power amplifiers and restoring arrangements of various types are shown in Figs. 4 and 5, while Figs. 6 and 7 show a number of pilot valves and servo-control units, as well as controllers varying the reference datum ψ.

4. CONTROLLER ELEMENTS

Each controller element is characterised by the variation of its output quantity φ_a as a function of the input quantity φ_e. These two quantities are related to one another by various equations:—

*A system of n network elements can be represented by n equations of the type

$$\ldots m_{k1}\,x_1'' + r_{k1}\,x_1' + c_{k1}\,x_1 + \ldots + m_{k2}\,x_2'' + r_{k2}\,x_2' + c_{k2}\,x_2 + \ldots + c_{kn}\,x_n = 0$$

($k = 1, 2, \ldots, n$). The elements are non-interacting if $c_{jk} \neq c_{kj}$, $r_{jk} \neq r_{kj}$, and $m_{jk} \neq m_{kj}$. The system is a coupled system if these coefficients are equal.

†Only systems operating as under (1) are treated in the following lines; systems based on methods (2-5) are discussed in the book "Continuous control processes" (Stetige Regelvorgänge) also written by the author.

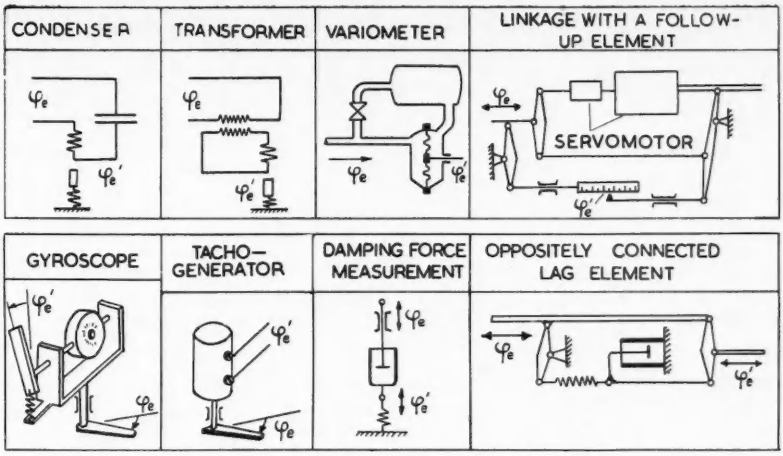

Fig. 3.
Differentiating elements.

(a) differential equation

$$f_a(\varphi_a, \varphi_a', \varphi_a'', \ldots) = f_e(\varphi_e, \varphi_e', \varphi_e'', \ldots) \quad (1)$$

(b) transfer function or frequency characteristic

$$\boldsymbol{F}(i\omega) = \varphi_a / \varphi_e \quad \ldots \quad (2)$$

if $\varphi_e = \varphi_{eo}\, e^{i\omega t}$, and $\varphi_a = \varphi_{ao}\, e^{i(\omega t + \alpha)}$ then the transfer function will be

$$\boldsymbol{F}(i\omega) = \varphi_a / \varphi_e = \varphi_{ao}\, e^{i\alpha} / \varphi_{eo} \quad \ldots \quad (3)$$

while the magnification is defined as

$$V = \left[\frac{\varphi_a}{\varphi_e} \right]_{\omega \to 0} \quad \ldots \quad (4)$$

(c) the transient function $\varphi(t)$, which is the solution of the differential equation (1) for the particular case that the input is varied suddenly to a value which is one unit higher, by a so-called " unit step disturbance."

The relations between the various characteristic quantities are

$$\varphi_a^{(n)} = (i\omega)^n\, \varphi_a, \quad \varphi_e^{(n)} = (i\omega)^n\, \varphi_e \quad \ldots \quad (5)$$

$$\boldsymbol{F}(i\omega) = \varphi_{ao}\, e^{i\alpha} / \varphi_{eo} = f_a((i\omega), (i\omega)^2, \ldots) / f_e((i\omega), (i\omega)^2, \ldots) \ldots \quad (6)$$

According to Fourier's theorem, any function can be represented by the sum of its harmonic components, i.e., by its amplitude-frequency spectrum. The Fourier expression for a unit step disturbance is

$$\varphi_e(t) = \frac{1}{2\pi i} \int_{-\infty}^{+\infty} d\omega\, \frac{e^{i\omega t}}{\omega} \quad \ldots \quad (7)$$

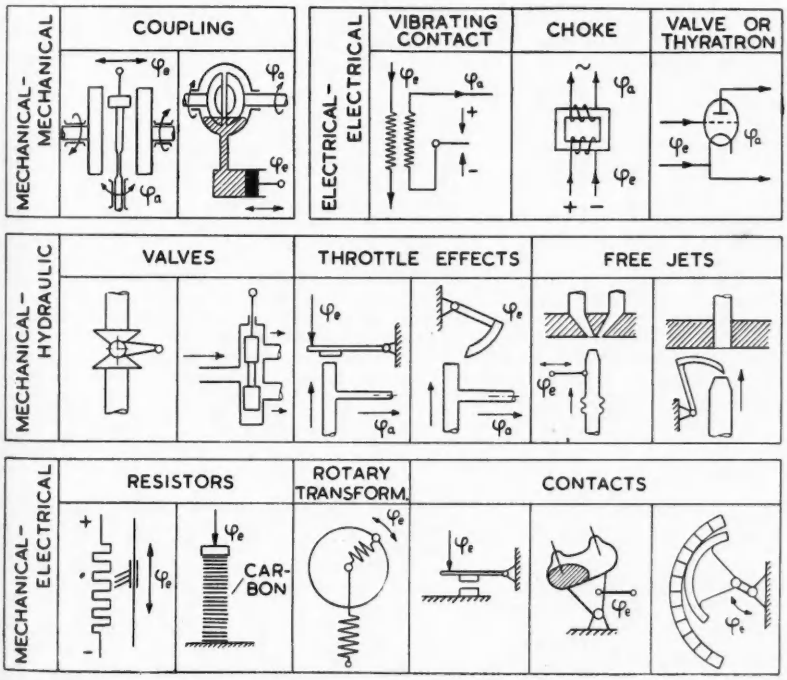

Fig. 4.
Power switches and power amplifiers.

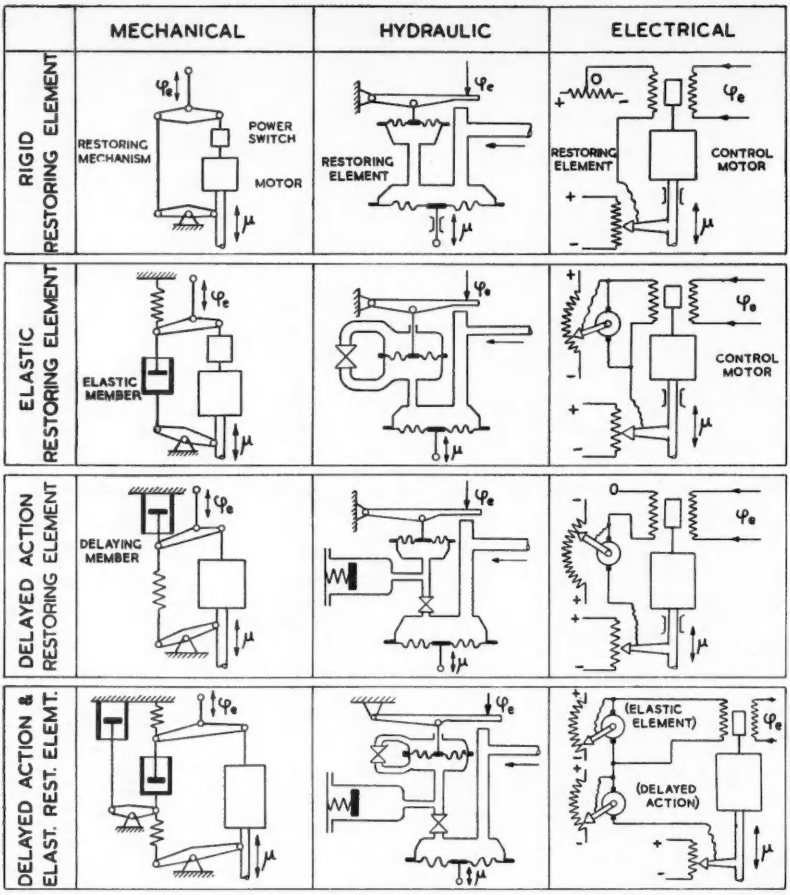

Fig. 5.
Restoring devices.

and since the output function is $\varphi_a = \boldsymbol{F}(i\omega)\ \varphi_e$, we have

$$\varphi_a(t) = \frac{\varphi_{eo}}{2\pi i} \int_{-\infty}^{+\infty} d\omega\ \boldsymbol{F}(i\omega)\ \frac{e^{i\omega t}}{\omega} \quad .. \quad (8)$$

Eq. (8) is, therefore, the equation for the transient function defined under sub-section (c). The above equations can also be expressed in the real domain, for instance eqs. (7) and (8) then become

$$\varphi_e(t) = \frac{1}{2} + \frac{1}{\pi} \int_0^{\infty} d\omega\ \frac{\sin \omega t}{\omega} \quad .. \quad .. \quad (7')$$

$$\varphi_a(t) = \frac{1}{2}(\varphi_{ao})_{\omega \to 0} + \frac{1}{\pi} \int_0^{\infty} d\omega\,(\varphi_{ao})_{\omega=\omega}\ \frac{\sin(\omega t + \alpha(t))}{\omega} \quad .. \quad (8')$$

The unit step disturbance of the input shown in Fig. 8a causes a variation of the output quantity as indicated in Fig. 8b. In the complex plane, the unit input disturbance is a vector φ_e along the real axis. The output quantity φ_a is then represented by the locus of the tips of the φ_a vectors for various frequencies ω and phase angles α, determined by means of eq. (8). The inverse locus (Fig. 9b) determines $\varphi_e(t)$ when the output φ_a is a unit step function. For any point of the input function (Fig. 9a), the corresponding point of the inverse locus is obtained by plotting as a negative angle the angle between the original φ_e vector and the origin, and taking for the length of the inverse vector φ_e the reciprocal of the length of the original φ_e vector. This can also be performed mechanically by means of the inversors of Peaucellier and Hart.

The curve of the transient function $\varphi(t)$ is related to the locus of the transfer function $\boldsymbol{F}(i\omega)$. When $\varphi(t)$ tends asymptotically towards its maximum amplitude for $t \to \infty$, the initial point of the $\boldsymbol{F}(i\omega)$ locus is determined for the frequency $\omega = 0$. When the initial variation of $\varphi(t)$ occurs for $t = 0$, the end point of the transfer function $\boldsymbol{F}(i\omega)$ is obtained for $\omega \to \infty$ (see Figs. 10a and 10b).

If the controller element is characterised by the time-constants T_i, and the input-control coefficients r_n (in general, $i \leq 4$, and $-1 \leq n \leq +1$), it can be determined by a differential equation of the type

$$T_i\,\varphi_a^{(i)} + \ldots + T_1\,\varphi_a' + \varphi_a = \ldots r_{-1} \int \varphi_e\,dt + r_o\,\varphi_e + r_1\,\varphi_e' + \ldots \quad .. \quad (9)$$

The transfer function is then

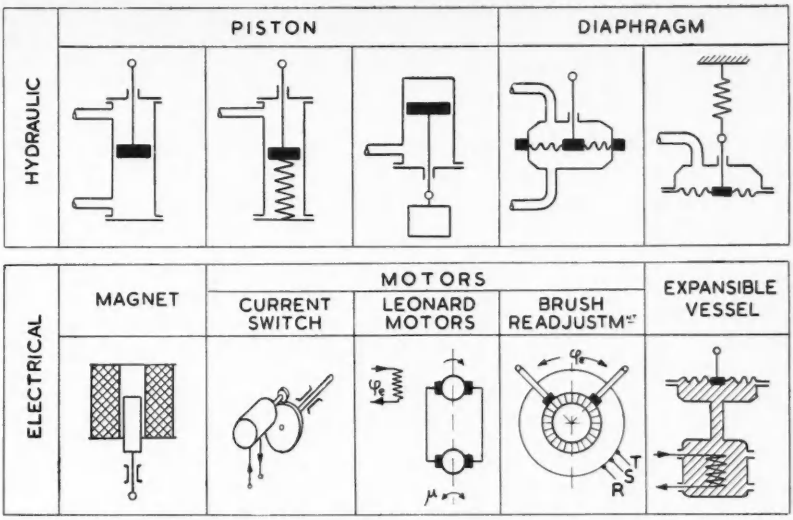

Fig. 6. **Pilot valves and servo-control units.**

$$\mathbf{F}(i\omega) = \frac{\ldots + r_{-1}(i\omega)^{-1} + r_o + r_1(i\omega) + \ldots}{1 + T_1(i\omega) + T_2(i\omega)^2 + \ldots + T_i(i\omega)^i} \quad (10)$$

A pure lag network is, therefore, characterised by the equations

$$T_i\,\varphi_a^{(i)} + \ldots + T_2\,\varphi_a'' + T_1\,\varphi_a' + \varphi_a = r_o\,\varphi_e \quad (11)$$

and $$\mathbf{F}(i\omega) = r_o/[1 + T_1(i\omega) + \ldots + T_i(i\omega)^i] \quad (12)$$

while a network with input-control devices and no lag terms is determined by

$$\varphi_a = \ldots r_{-1}\int \varphi_e\,dt + r_o\,\varphi_e + r_1\,\varphi_e' + \ldots \quad (13)$$

and $$\mathbf{F}(i\omega) = \ldots r_{-1}(i\omega)^{-1} + r_o + r_1(i\omega) + \ldots \quad (14)$$

The amplitude and the phase angle are obtained from eqs. (10), (12), or (14), in the usual manner, i.e., by determining

$$|\,\mathbf{F}(i\omega)\,|^2 = |\,A + iB\,|^2 = |\,\varphi_a/\varphi_e\,|^2 = A^2 + B^2,$$
$$\text{and,}\quad \tan\alpha = B/A.$$

In differential equation (9), the terms of the first order represent aperiodic-decay phenomena, the second-order terms determine damped oscillations while the third and higher-order terms can be related to oscillations of increasing or decreasing amplitude.

Lag Networks: According to eq. (12) all loci $\mathbf{F}(i\omega)$ of lag networks must begin and end on a straight line parallel to one of the co-ordinate axes. For when $\omega \to \infty$, the highest-order term in the polynomial of the numerator or the denominator will always be preponderant, so that the vector will be parallel to one of the co-ordinate axes, while for an initial frequency equal to zero, similarly, the lowest-order term will be preponderant.

All loci of lag control networks begin at the point r_o on the real axis and continue in the region of negative phase angles. The locus of the first-order term is a semi-circle, which expands when a second-order term is added (owing to resonance), and the locus with a third-order term may burst open at a pole (possible instability). The fourth-order locus shows two maximum values for amplitude. When $\omega \to \infty$ the loci approach the zero point, with a phase angle of $-90°$ for first-order curves, $-180°$ for second-order curves, $-270°$ for third-order curves, etc. Stability conditions will be discussed later. In the case of a second-order curve corresponding to $T_2\,\varphi'' + T_1\,\varphi' + \varphi = 0$, the intersection of the locus with the imaginary axis gives the natural frequency and the time constants: $\omega = \omega_o = 1/\sqrt{T_2}$ = natural frequency of the undamped system, $|\,\varphi_a\,|\,/\,|\,\varphi_e\,| = 1/(\omega_o\,T_1)$, while the damping D of the system is given by $D = T_1/(2\sqrt{T_2})$. Further details regarding the damping factor D are given in Section 12 (d).

Input-Control Networks: The derivatives of the input quantity (eq. (13)) serve to generate phase leads which damp out oscillations in the control circuit. The integral terms neutralise any permanent control deviations when the state of equilibrium is disturbed. The steady state can only exist under conditions conforming to the pre-set reference datum, as under other conditions the integral terms are time-variant. Generally, the expressions do not include more than the second

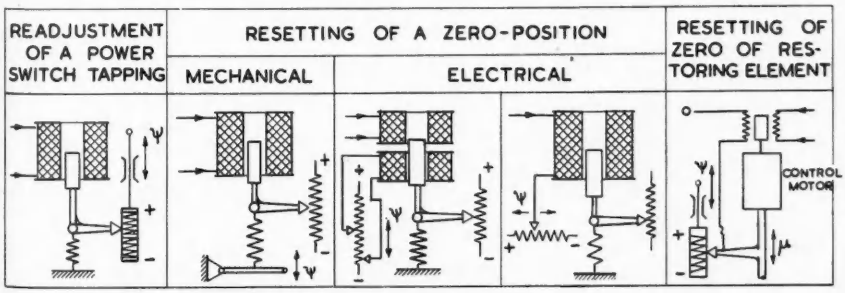

Fig. 7. **Controller elements for varying the reference datum.**

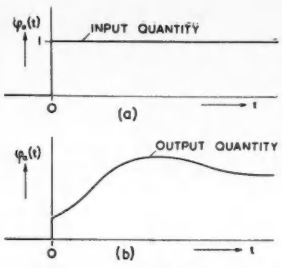

Fig. 8. **Transient functions.**

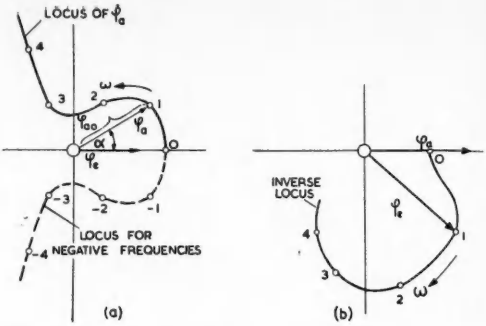

Fig. 9. **Transfer function loci.**

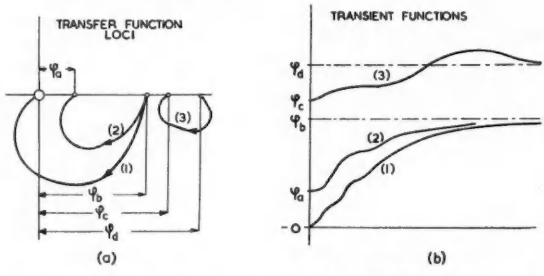

Fig. 10. **Relationship between transfer locus and transient function.**

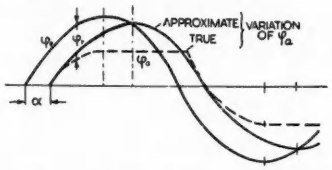

Fig. 11. **Effect of friction. Approximation of real variation by a more easily calculated sine curve.**

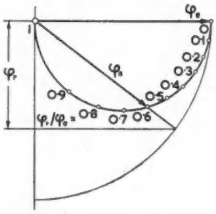

Fig. 12. **Amplitude-variant locus of transmission element with friction.**

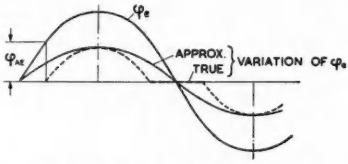

Fig. 14. **Approximation of curve with loss in response sensitivity by means of a sine curve.**

derivative and the double integral terms. The loci of the input quantity derivatives are straight lines emanating from the origin, while the integral curves are straight lines approaching the origin and coming from infinity. The transient functions have poles at the derivatives.

The loci begin and end at points determined by omitting certain terms from their equations. For instance, the origin of the locus (i.e., the end-point of the transient function) is obtained by omitting all terms with derivatives, since in this stationary state all the derivatives are zero. Thus, for instance, the origin of the $\mathbf{F}(i\omega)$ locus corresponding to $T_2\,\varphi_a'' + T_1\,\varphi_a' + \varphi_a = r_0\,\varphi_e + r_1\,\varphi_e' + r_2\,\varphi_e''$, is given by $\varphi_a = r_0\,\varphi_e$. At the end-point of the locus (origin of the transient function) the highest-order derivative prevails, so that all others may be neglected; therefore, $T_2\,\varphi_a'' = r_2\,\varphi_e''$, which after integrating twice, gives $T_2\,\varphi_a = r_2\,\varphi_e$.

Transmission Time : The time-lag in the transmission of commands within the control system can also be represented by a frequency-dependent function, such as

$$\varphi_a / \varphi_e = \mathbf{F}(i\omega) = e^{-i\omega T_l}$$

where T_l = transmission delay ($T_l = l\sqrt{LC}$ in the case of a telegraph wire of length l). Then $\varphi_a(t) = 0$ for $t < T_l$, and $\varphi_a = \varphi_e(t - T_l)$ when $t \geq T_l$. Generally, transmission delay occurs in conjunction with

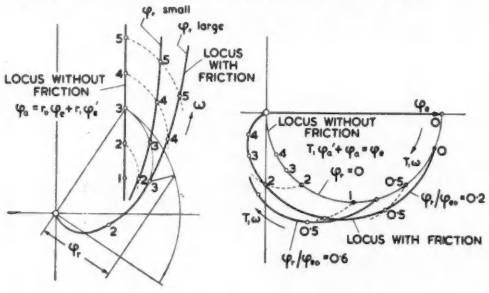

Fig. 13. **Families of curves of control devices with a zero-lag element with friction.**

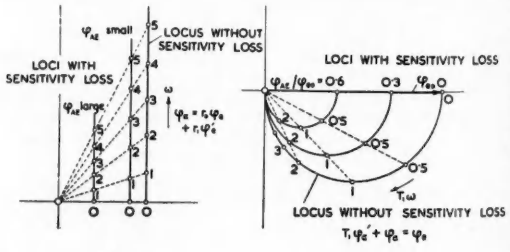

Fig. 15. **Families of curves for control devices with sensitivity losses.**

some distortion, transformation, or phase-variation in the controller elements. The transient functions with a transmission time lag are the same curves as originally but displaced over distances corresponding to the time lags considered.

Effect of Friction: In the controller element with friction, the input quantity must first reach a value equal to that of the negative friction before the output is affected. The output curve obtained from a sinusoidal input with friction is shown in Fig. 11. The output remains constant in the neighbourhood of the point of inflexion of the input quantity, and decreases again in amplitude as soon as the input is lower than the frictional amplitude difference. In approximating the behaviour of the controller element with friction, the discontinuous output quantity is replaced by a sinusoidal curve, with a phase angle α determined by $\sin\alpha = \varphi_r/\varphi_{eo}$, where φ_r = frictional amplitude. The approximation curve with this phase-lag is assumed to be generated by a lag-element of the first order, with the output amplitude

$$\varphi_{ao} = \varphi_{eo} \cos \alpha \quad .. \quad .. \quad (15)$$

hence the transfer function is

$$\varphi_a/\varphi_e = \mathbf{F}_r(i\omega) = \sqrt{1-(\varphi_r/\varphi_{eo})^2}\, e^{i \arcsin \varphi_r/\varphi_{eo}} \quad (16)$$

The plot of this transfer locus is a semi-circle which depends on the amplitude-ratio and not on frequency. Greater phase-shifts are obtained with smaller amplitudes. Friction generally occurs in conjunction with other properties of the controller elements. Instead of a single locus contour, it is then necessary to plot families of curves corresponding to different friction values and also varying with frequency (see Figs. 12 and 13).

Response Sensitivity: Response sensitivity is characterised by a decrease in amplitude without a phase shift. If φ_{AE} denotes the sensitivity loss, then the relation between the input and output quantities φ_{eo} and φ_{ao} is

$$\varphi_{ao} = \varphi_{eo} - \varphi_{AE} \quad .. \quad .. \quad (17)$$

so that we can write

$$\mathbf{F}_{AE}(i\omega) = \varphi_a/\varphi_e = 1-(\varphi_{AE}/\varphi_{eo}) \quad .. \quad (18)$$

The transfer locus again depends on amplitude and not on frequency (Figs. 14 and 15). In control systems, the same element can generate a sensitivity loss or a frictional loss according to its position in the system. For instance, a certain amount of clearance between a pin and fork link will cause initially a sensitivity loss, which can become a frictional loss if the pin is pressed against the fork component by a retaining spring.

(To be continued.)

ITALY

Assessment of the Resistance of Refractory Materials to Sudden Temperature Variations

By A. GIANNONE. (From *Il Calore*, Vol. 19, No. 11, November, 1948, pp. 398-401, 6 illustrations.)

AN important factor in the design and operation of furnaces is the resistance of refractory materials to sudden temperature variations. Many investigations of this problem have been undertaken with a view to establishing a criterion for the assessment of this factor, which is, in fact, the result of a number of variables which mutually affect each other, such as thermal expansion and thermal conductivity, elasticity and mechanical strength, constraining conditions, etc. These properties, in turn, depend on the chemical composition of the material, and the manufacturing process used, and, therefore, on the texture and porosity of the material.

The theoretical considerations related to the calculation of stresses occurring in a refractory layer subjected to a given thermal cycle, and the effect of the various factors on the stability of the material, have been the subject of previous articles by the author[1]. The present article will be confined to the development of a graphical method for the determination of the stresses acting in a plane wall of infinite length. For this purpose it is necessary to know:—

(1) the temperatures occurring at various distances within the wall at a given instant, i.e., the temperature diagram $T = T(x)$, for $t = t_0$ = constant;
(2) the mean value of the coefficient of thermal expansion α; if the material has anomalous expansion characteristics, as is usually the case with refractory silica-type materials, it is necessary to know the entire curve relating strain to the temperature of the material:

$$\epsilon = \Delta l/l = \epsilon(T);$$

(3) the modulous of elasticity E, and
(4) the rupture strength R of the material.

If α can be regarded as constant, the ordinates of the temperature diagram, multiplied by α, will give the non-elastic expansions, i.e., the expansions which would occur at each point as a result of the temperature variation, so that, in general, the diagram

$$\epsilon = \epsilon(x)$$

will be obtained point by point, by eliminating the temperature T between the diagram giving $T = T(x)$ and the thermal expansion curve $\epsilon = \epsilon(T)$, as shown in Fig. 1. As the material considered in this figure is of the silica type, one can observe the anomalies due to the transformations of tridymite (130° C), cristobalite (210° C) and quartz (575° C). The ordinates of diagram (b), multiplied by the elasticity modulus E, give the non-elastic stresses. Even if the wall is assumed to be completely free to expand and rotate, we still have to retain the assumption that plane cross-sections remain plane; therefore, elastic stresses will be generated which will tend to compensate, at least in part, the more or less irregular deformations due to thermal action. Consequently, with the construction shown in Fig. 1b, the " compensation line " of the diagram, i.e., the straight line PQ, will be determined. This line is drawn in such a way that the vertical line to the centre of gravity of the trapezium $APQB$ will coincide with that of the area $AA'B'B$. The elastic stresses will then be represented by the ordinates, with the plus or minus sign reversed, comprised between the original diagram $A'B'$ and the line PQ, as shown in Fig. 1c.

If the wall is free to expand but not allowed to rotate, it will be sufficient to draw the compensation line parallel to the X-axis (Fig. 2). If, however, it is rigidly held so that it can neither expand nor rotate, then the same diagram $A'B'$, with the signs reversed, will constitute the stress diagram (Fig. 3). In both these cases, as might have been expected, the stresses are greater than in the case of the unconstrained wall.

From the above treatment it is apparent that the absolute values of the stresses will be greater, the steeper the temperature curve, and the more it diverges from the straight line corresponding to stationary con-

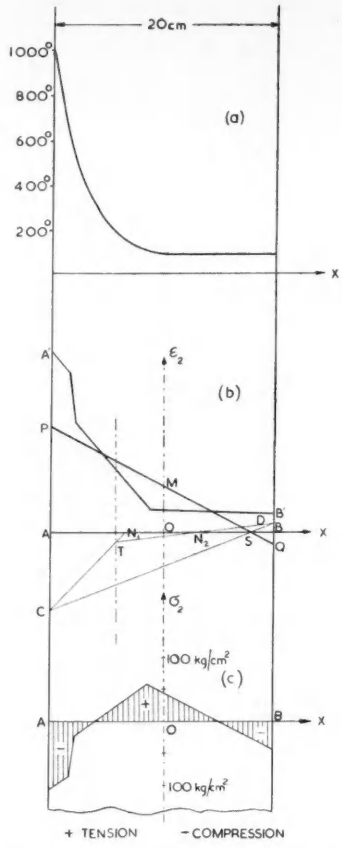

Fig. 1. **Wall of silica-type refractory material, assumed to be of infinite length, completely free to rotate and expand.**
1 (a) Temperature diagram, assuming the temperature of one wall surface has been raised to 1000° C.
1 (b) Thermal expansion diagram, and construction of the compensation line PQ.
1 (c) Diagram of elastic stresses.

ditions; but it should not be concluded that internal stresses do not arise under stationary conditions. In fact, considering one of the refractory bricks taken as an element of a large vertical wall, it is apparent that it is not generally free to rotate and assume the curved shape which it should have according to the temperature diagram, but that, at most, it will be able to expand uniformly with the entire wall. Therefore, after having drawn the straight-line temperature diagram $A'B'$ (Fig. 4) it will be sufficient to draw the horizontal compensation line. The ordinates included between $A'B'$ and this line, after changing their signs as previously, and assuming α and E to be constant, will give the values of the prevailing stresses. Still higher stresses will be obtained if the material has anomalous expansion characteristics.

In the case of sudden cooling, the temperature and stress diagrams will be curves with a shape similar to that of Fig. 5. The stresses can reach high peak values, and can produce scaling or even the rupture of the material.

According to our criteria, the stresses occurring in refractory layers are essentially normal stresses, giving rise to more or less regular bending deflections. The

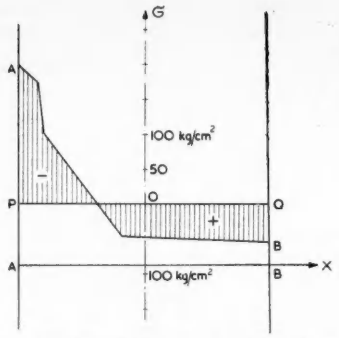

Fig. 2. **The same wall as in Fig. 1, assumed to be free to expand but without rotating.**

distribution of the internal stresses is a one-dimensional distribution, depending only on x, the distance of the layer being considered from one of the faces of the wall. Only in the case of isolated bricks, or in the vicinity of the edges of the walls, that is, when the isothermal surfaces are no longer plane parallel surfaces, will it happen that the temperature T and the stresses σ will also be functions of the other co-ordinates of the system. Therefore, according to the theory of structures, tangential stresses τ will appear, which are very

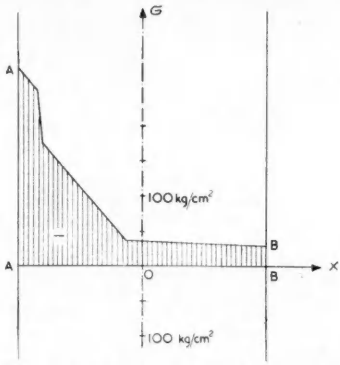

Fig. 3. **The same wall as in the previous figures, assuming rigid constraint, so that the wall can neither rotate nor expand.**

dangerous in view of the low resistance of refractory materials to stresses of this kind. Near the edges, the isothermal surfaces become more and more curved, and the tangential stresses assume high values until they produce characteristic fractures with an inclination of approximately 45 degrees, which correspond to the principal directions of pure shear stresses[2].

In order to establish a criterion for assessing the resistance of a given structure to sudden temperature changes, following a reasoning similar to that applied in all stability calculations, it will be sufficient to consider the ratio of the maximum stress σ_{max} to the rupture strength R:—

$$\frac{1}{S} = \frac{\sigma_{max}}{R}$$

where S is the modulus of resistance to sudden temperature changes, and in general, $S > 1$. This modulus is, of course, related to a given thermal cycle, that is, to a given rate of heating and cooling. Furthermore, it should be borne in mind that the tensile stresses will generally be the more dangerous stresses, since they

correspond to a lower value of the rupture strength modulus R.

Without investigating in detail the individual factors which determine the value of S, we shall, for the time being, particularly consider the effect of thermal conductivity.

The theoretical determination of the function $T = T(x, t)$ is never a simple matter, and frequently it is even impossible. After overcoming considerable difficulties in the calculation and assuming particular boundary conditions which do not always correspond to reality, we obtain functions depending not only on the thermal conductivity K, but also on the specific heat c and the specific gravity γ. These variables are not easily determined with normal industrial measuring equipment; moreover, they vary with temperature and, strictly speaking, assume different values at different points of the wall thickness. Many methods have been suggested for the determination of the thermal conductivity of refractory materials[3], but it is evident from the foregoing considerations that, as regards the resistance of the material to sudden temperature variations, the value of K need only be known approximately, whereas it is essential to determine the diagram of the temperature distribution within the wall at various instants of the thermal cycle. To appreciate the importance of this diagram, it will be sufficient to observe that, if we consider the extreme case of a wall rigidly constrained on all sides, and assume constant values for E and α, this diagram will also represent the actual stress diagram of the wall on a convenient scale.

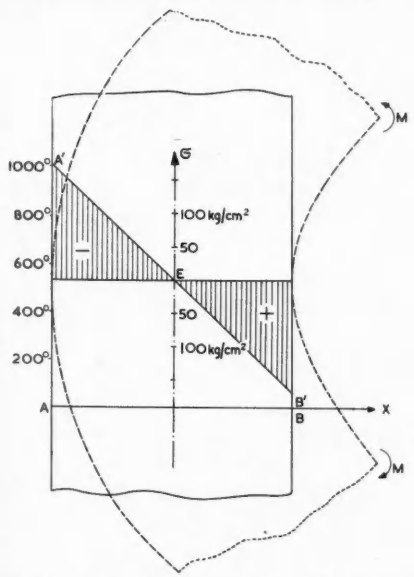

Fig. 4. **Plane wall, of infinite length, capable of expanding without rotating. Constant temperature distribution and constant coefficient of expansion and modulus of elasticity.**
$\alpha = 3{\cdot}5 \times 10^{-6}$; $E = 10^5$ kg/m².

On the basis of the above considerations, it will be possible to assess the behaviour of a given type of refractory material by making the following test:—

A refractory panel, which reproduces as closely as possible the type of lining required, is placed in front of the opening of a furnace available for tests (Fig. 6). A vertical furnace is most convenient, since it enables the panel to be laid down horizontally. In this way, a more uniform distribution of temperature will be achieved. A number of thermocouples are inserted

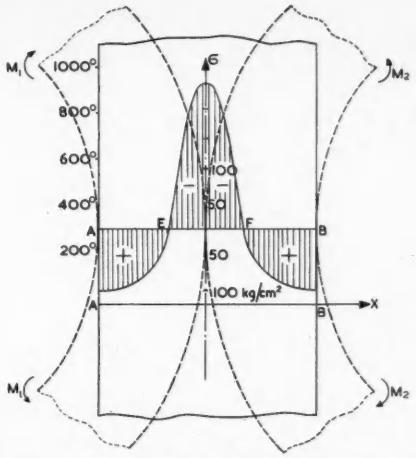

Fig. 5. **Plane wall, of infinite length, suddenly cooled on both outer surfaces, assumed to be free to expand but without rotating.**
$\alpha = 3{\cdot}5 \times 10^{-6}$; $E = 10^5$ kg/cm².

through appropriate passages inside the test panel, along the axis AB of the furnace. The temperature is then raised as uniformly as possible, the temperature diagram is recorded at regular time intervals, and the corresponding internal stresses are calculated for each set of readings. The same procedure is then repeated for the cooling process. By varying the rate of temperature rise or cooling, it will then be possible to determine the limit rate of temperature change beyond which the stresses become excessive and the modulus of resistance to temperature variations, S, approaches unity.

The method of testing refractory materials in the form of panels instead of individual bricks is now generally accepted[4] as yielding the most satisfactory results; it also has the advantage of more closely simulating the actual working conditions and does not require special precautions for thermal insulation, since, from considerations of symmetry it is obvious

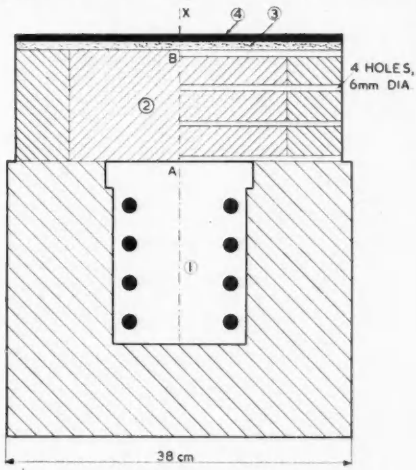

Fig. 6.
1 = Electric furnace with silit rods.
2 = Refractory panel being tested.
3 = Insulating material.
4 = Metal cover.

that in the central part of the panel the isothermal surfaces are plane and parallel to the outer faces, a condition which is necessary for the application of the suggested arrangement. Furthermore, it is not necessary to undertake calorimetric measurements, since temperature measurements only are required.

In checking the behaviour of thermal plants already in operation, the thermocouples should be placed at the most important points of the system (curved surfaces, wall centres, and edges) in order to determine the temperature diagram and the corresponding stress distribution within the layer, while the furnace is in operation. After finding the maximum permissible stresses, a spacial temperature regulation sheet for the operation of the furnace can be established, which will enable the furnace to work within the safety and endurance limits of the refractory material used.

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SWEDEN

Effects of Sulphur Content and Dewpoint of Flue Gases upon Boiler Operation

By S. JUHÁSZ. (From *Teknisk Tidskrift*, Vol. 79, No. 4, January 22, 1949, pp. 49-56, 14 illustrations.)

AMONG the various factors influencing the dewpoint of flue gas, the effects of the sulphur content are the most important. For a flue gas containing sulphur, the dewpoint is defined as the highest temperature of a surface at which the vapour particles (water vapour and sulphuric acid) are in equilibrium with the liquid particles. This temperature constitutes the so-called effective dewpoint. The problems involved were first investigated by Johnstone some twenty years ago, and even to-day his experiments retain their original importance.

According to Johnstone, the presence of minute quantities of sulphur is sufficient to raise the dewpoint. This rise in dewpoint may be as much as 60° C, depending upon the amount of sulphur present. The theoretical dewpoint is easily determined by computation, but the effective dewpoint can be ascertained only by actual measurement. Various types of instruments for the measurement of the dewpoint are used in laboratory tests, but their employment has not become so widespread as that of other steam plant instruments. This is probably due to the fact that none of these laboratory test instruments satisfies the requirements of plant practice. Moreover, in the instruments hitherto available, the dewpoint indication is obtained by artificially producing condensation below or in the vicinity of the dewpoint. These instruments do not, therefore, indicate the true conditions obtaining in the boiler, where condensation may take place at various points.

The author, in collaboration with P. Fonó, has developed a dewpoint meter (patents applied for) consisting of a long tube, in which condensation takes place, and containing a number of thermocouples distributed over the length of the tube. As soon as condensation takes place, the thermocouple nearest the point of condensation is automatically actuated. The only precaution to be taken in the operation of this instrument consists in ensuring that condensation actually occurs within the tube. An apparatus of this type is diagrammatically represented in Fig. 1 which also indicates the electrical circuits. This particular instrument is fitted with three measuring points, but a larger number may be provided, thus increasing accuracy of measurement.

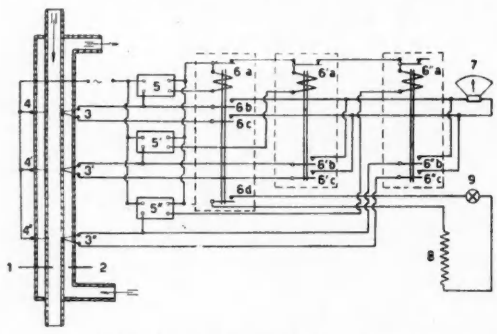
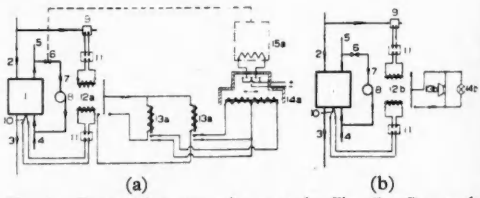


Fig. 1. **Proposed automatic dewpoint meter.**

(1) Glass tube; (2) cooling water; (3) thermocouple; (4) electrodes; (5) amplifier; (6) relays; (7) millivoltmeter; (8) heating element; (9) light signal.



(a) (b)

Fig. 2a. **Proposed automatic control.** Fig. 2b. **Proposed automatic indication of danger temperature in air preheater operation.**

(1) Air preheater; (2) flue gas entering; (3) flue gas leaving; (4) cold air; (5) preheated air; (6) hot air recirculation damper; (7) hot air recirculation duct; (8) hot air recirculation fan; (9) dewpoint indicator; (10) metal temperature indicator; (11) amplifier; (12) differential relay; (13a) relay; (13b) acoustic signal; (14a) magnetic switch; (14b) optical signal device; (15a) electric motor for damper control.

In order to prevent the fall of the gas temperature to or below the dewpoint in air preheater operation, recirculation of the gases or of the preheated air is employed. The realisation of these methods, however, also requires measurement of the metal temperature at the most endangered point in order to actuate the control mechanism adjusting the dampers, etc. An automatic controller (patent applied for) developed by the author in collaboration with T. Laczay is outlined in Figs. 2a and 2b. The apparatus shown in Fig. 2a is fully automatic and provides continuous control, while the somewhat simpler apparatus outlined in Fig. 2b operates an optical signal when the actual metal temperature approaches the dewpoint within 30° C.

Rheological Criteria for Two-Stroke Cycle Scavenge Systems

By E. NIEDERMAYER. (From *Maschinenbau und Wärmewirtschaft*, Vol. 4, Nos. 1 and 2, January and February, 1949, pp. 1-7 and 18-24, 11 illustrations)

(*Continued from May issue*)

2. FOR THE RHEOLOGICAL PHENOMENA OCCURRING AFTER INLET SLOT OPENING, during which part of the charge flows into the scavenge air chamber, we can write on the assumption of a constant cylinder volume:

$$-d\left(\frac{1}{v_z}\right)=\left[\mu_a f_a W_a\,dz+\mu_e f_e W_e\,dz\right]\Big/(\rho V_h) \quad (17)$$

where $W_a = w_a/v_{ma}$, $W_e = w_e/v_{me}$,

and $$\rho=\frac{\epsilon-\sigma_a}{\epsilon-1}-\frac{1}{2}(\sigma_e+\sigma')$$

so that

$$-\frac{1}{W_a}d\left(\frac{1}{v_z}\right)=\left[\frac{\sigma_a\,d\alpha}{K_a}+\frac{W_e\,\sigma_e\,d\alpha}{W_a K_e}\right]\Big/(45\rho).$$

This equation can be integrated if we replace $W_e/W_a = \varphi$ by a mean value regarded as constant.

The mean value of φ during the considered time interval from α_e to α', can be calculated for various values of p_{ze} and p_s for over-critical and under-critical conditions, by means of the equation

$$\varphi=\frac{1}{p_{ze}-p_s}\int_{p_{ze}}^{p_s} dp_z\,W_e/W_a.$$

The results are given in Fig. 6, where φ_{mean} is shown as a function of the pressure-ratio p_{ze}/p_s for various values of p_s/p_a.

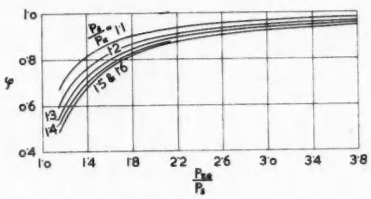


Fig. 6. **Value of φ.**

Using the abbreviations

$$Z_a=\int_{\alpha_e}^{\alpha'} d\alpha\;\sigma_a \quad\text{and}\quad Z_e=\int_{\alpha_e}^{\alpha'} d\alpha\;\sigma_e,$$

the integration of eq. (17) thus gives, approximately,

(a) for over-critical flow: $p_{ze} \geq 1{\cdot}85\,p_a$:

$$\frac{\sqrt{T_{za}}}{\rho}\left[\frac{Z_a}{K_a}+\frac{\varphi Z_e}{K_e}\right]$$
$$=22{\cdot}8\left[\left(\frac{P}{1{\cdot}85}\right)^{0{\cdot}128}-Q^{0{\cdot}128}\right]$$
$$+3{\cdot}24\,P^{0{\cdot}128}\left[0{\cdot}924-2\sqrt{\psi}\left(\frac{2}{3}\psi+1\right)\right]$$
$$=B_1'+B_1''=B_1, \quad .. \quad .. \quad .. \quad (18)$$

where

$\psi=(p_s/p_a)^{0{\cdot}256}-1$, $P=p_{za}/p_a$, and $Q=p_{za}/p_{ze}$;

(b) for under-critical flow: $p_{ze} < 1{\cdot}85\,p_a$:

$$\frac{\sqrt{T_{za}}}{\rho}\left[\frac{S_a}{K_a}+\frac{\varphi S_e}{K_e}\right]$$
$$=3{\cdot}24\,P^{0{\cdot}128}\left[2\sqrt{\psi'}\left(\frac{2}{3}\psi'+1\right)-2\sqrt{\psi}\left(\frac{2}{3}\psi+1\right)\right]$$
$$=B_1''-A_2'' \quad .. \quad .. \quad .. \quad .. \quad .. \quad (19)$$

The values of B_1' and B_1'' as functions of $P = p_{za}/p_a$ are also shown in Fig. 4. (See May issue, p. 156).

The crank-angle α' at which the scavenging process commences, and which is also the upper limit of the integrals Z_a and Z_e, can now be determined with the aid of Fig. 5, as will be shown in the numerical example given under sub-section (B) 7.

3. TIME-AREA INTEGRAL FOR THE INLET SLOTS.

During the period determined by α' and $-\alpha_e$, a quantity of gas $S + \Delta S$ must flow through the inlet slots. This gas volume is expressed as a proportion of the stroke volume and referred to prevailing external conditions. The time-area integral calculation is simplified if the integration limits are made symmetrical with respect to the bottom dead centre. Thus, instead of the true slot height σ_e we introduce an equivalent inlet slot height σ_e' defined by

$$Z_e'=\int_{+\alpha}^{-\alpha} d\alpha\;\sigma_e'=\int_{\alpha'}^{-\alpha_e} d\alpha\,\sigma_e \quad .. \quad .. \quad (20)$$

The relationship is shown schematically in Fig. 7, and represented by the curves in Fig. 8.

Using eq. (11), the required time-area integral can now be determined as

$$Z_e'=45\,K_e(S+\Delta S)\Big/\sqrt{2gH_{ad}\,T_a/T_e} \quad (21)$$

Having calculated the height of the inlet slots by means of eq. (20), the quantity

$$K_e(1-0{\cdot}5\,\lambda)(S+\Delta S)\Big/155$$

can be determined from Fig. 9, and, therefore, the value of the sum $S + \Delta S$ is also determined.

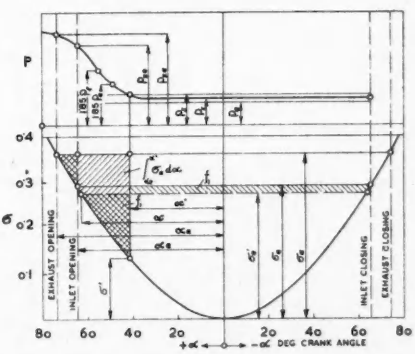


Fig. 7. **Schematic representation of time-area integrals for inlet and exhaust slots and variation of the charging pressure.**

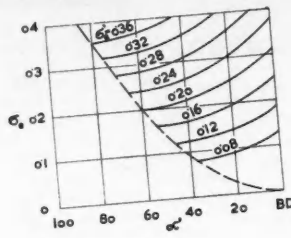


Fig. 8. **Relationship between** σ_e, σ_e' and α', for $\lambda = 1/4$

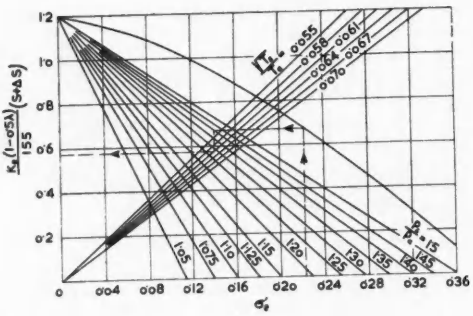


Fig. 9. **Nomogram for the determination of the air consumption factor, valid for values of** λ from 1/3 to 1/5.

4. Calculation of the Air Consumption Factor.

The quantity of residual gas which has come into the scavenge air chamber is given by

$$\Delta S\,(1+\tau)\,V_h\,p_a/T_a$$

$$= V_h\left[\frac{p_{ze}}{T_{ze}}(1-\sigma_e+\delta)-\frac{p_s}{T_z'}(1-\sigma'+\delta)\right]$$

where $\delta = (1-\sigma_a)/(\epsilon-1)$.

In this relation, τ is the ratio of the weight of gas flowing into the exhaust gas manifold, during the period from α_e to α', to the weight of the residual gas entering the scavenge air chamber during the same period of time:—

$$\tau = \int dz\,\mu_a f_a W_a \Big/ \left[\int d_z\,\mu_e f_e W_e\right].$$

Assuming $W_a = w_a/v_{ma}$ = constant, $W_e = w_e/v_{me}$ = constant, and hence $W_a/W_e = 1/\varphi$ = constant, we have

$$\tau \simeq K_e Z_a \big/ (\varphi K_a Z_e) \quad .. \quad .. \quad (22)$$

so that

$$\Delta S = \frac{1}{1+\tau}\left[\frac{p_{ze}T_a}{p_a T_{ze}}(1-\sigma_e+\delta) - \frac{p_s T_a}{p_a T_z'}(1-\sigma'+\delta)\right] \quad (23)$$

where $T_z' = T_{za}\,(p_z/p_{za})^r$, $T_{ze} = T_{za}\,(p_{ze}/p_{za})^r$, and $r = (m-1)/m$. The required air consumption factor is, thus,

$$S = (S+\Delta S) - \Delta S.$$

5. Calculation of the Scavenge Ratio.

The scavenge ratio is reduced by the flow of residual gas into the scavenge air passages. The most unfavourable assumption is that complete mixing occurs with the scavenge air in the air chamber. Then according to eq. (2) for the case of negative-pressure scavenging:

$$L_z' = V_z p_z T_a\left[1-e^{-(S+\Delta S)C}\right]\Big/\left[p_a T_e'(S+\Delta S)\right]$$

where $C = V_h p_a T_e'/(V_z p_z T_a)$, and eq. (5) gives

$$\lambda_s' = L_z'\Big/\left[L_z' + \left(V_z - t_z'\frac{p_a T_e}{p_z T_a}\right)\frac{p_z T_a}{p_a T_{ze}}\right]. \quad (24)$$

In determining L_z', the temperature of the residual gas and scavenge air mixture can be taken as being approximately

$$T_e' = T_e\,[S + (\Delta S T_{ze}/T_e)]\Big/[S+\Delta S]$$

6. Discharge Coefficient.

The discharge coefficient is appreciably affected by the penetration of residual gases into the scavenge air chamber. The discharge coefficient μ measured on the engine under normal running conditions can be correlated to the discharge coefficient μ_M of the model by means of the equation

$$\mu = \mu_M\,S\,Z_e'\big/[S+\Delta S)\,Z_{eo}]$$

where $Z_{eo} = \int_{+a_e}^{-a_e} \sigma_e\,d\alpha.$

7. Numerical Example.

It is required to determine the optimum slot heights for a scavenge system with the scavenge-ratio curve shown in Fig. 1 for reverse-flow scavenging, the system being characterised by the following parameters:

$c = 9{\cdot}6$ m/sec, $\beta_a = 0{\cdot}172$, $\beta_e = 0{\cdot}30$, $\gamma = 50$ deg., $\mu_a = 0{\cdot}58$, $\mu_M = 0{\cdot}55$, $p_{ze}/p_a = 4$, $\epsilon = 16$, $\lambda = 0{\cdot}25$, $\lambda_v = 1{\cdot}8$, $\zeta = 1{\cdot}06$, $T_{za} = 1140°$ K, $T_a = 290°$ K, $T_s = 310°$ K, $b_e = 0{\cdot}180$ kg/PSh (1 PS = 0·98632 bhp).

In order to obtain the values giving the maximum engine power, the calculation is worked out for several values of σ_a and σ_e. In the following lines, the calculation is given for the slot heights $\sigma_a = 0{\cdot}36$ and $\sigma_e = 0{\cdot}24$. The constants are

$K_a = 90{\cdot}5$ (eq. (14)),
$K_e = 71{\cdot}7$ (eq. (12)), and
$\rho_a = 0{\cdot}747$ (eq. (13)).

Fig. 5 shows that $\int_{a_a}^{a_e} \sigma_a\,d\alpha = 0{\cdot}95$, hence, from eq. (15): $A_1 = 0{\cdot}474$. Furthermore, from Fig. 4: $1{\cdot}85\,p_{za}/p_{ze} = 2{\cdot}17$, so that $p_{ze} = 3{\cdot}42\,p_a$. From Fig. 3 we have: $p_s = 1{\cdot}4\,p_e$. According to Fig. 4: $B_1' = 1{\cdot}90$, $B_1'' = 1{\cdot}15$, $B_1 = 3{\cdot}05$. In addition, after employing eq. (17) to estimate $\sigma' = 0{\cdot}094$, we have $\rho = 0{\cdot}89$, and, therefore, $B_1 K_a \rho/\sqrt{T_{za}} = 7{\cdot}26$. From Fig. 6: $p_{za}/p_s = 3{\cdot}42/1{\cdot}40 = 2{\cdot}44$, $\varphi = 0{\cdot}905$, so that $\varphi\,K_a/K_e = 1{\cdot}14$. Further values are determined from Fig. 5, and are given in the following table:—

α'	Z_a	Z_e	$\varphi\,Z_e\,K_a/K_e$	$Z_a + \varphi\,Z_e\,K_a/K_e$
40°	5·05	2·05	2·34	7·39
45°	3·80	1·35	1·54	5·34

According to the previous calculation, the sum should be equal to 7·26. Interpolating, we thus obtain $\alpha' = 40°\ 15'$ and $\sigma' = 0{\cdot}091$ instead of the estimated value of 0·094. The true time-area integrals are, therefore: $Z_a = 5{\cdot}95 - 0{\cdot}95 = 5{\cdot}00$, and $Z_e = 2{\cdot}00$.

The temperatures are

$T_{ze} = 1140\,(3{\cdot}42/4{\cdot}00)^{0{\cdot}254} = 1095°$ K, and
$T_z' = 1140\,(1{\cdot}40/4{\cdot}00)^{0{\cdot}254} = 878°$ K.

The weight ratio, according to eq. (22) is: $\tau = 2{\cdot}20$, and from eq. (23): $\Delta S = 0{\cdot}089$.

From Fig. 8 we find that for $\sigma_e = 0{\cdot}24$ and $\alpha' = 40° 15'$ the equivalent inlet slot height is $\sigma_e' = 0{\cdot}222$.

From Fig. 9, for $T_a/T_a = 0{\cdot}061$ and $p_s/p_a = 1{\cdot}4$, we have:

$$K_e(1 - 0{\cdot}5\lambda)(S + \Delta S)/155 = 0{\cdot}575, \text{ so that}$$

$$S + \Delta S = 0{\cdot}575 \times 155/[71{\cdot}7\ (1 - 0{\cdot}125)] = 1{\cdot}425,$$

and hence the air consumption factor is:

$$S = 1{\cdot}425 - 0{\cdot}089 = 1{\cdot}336.$$

Scavenge Ratio: Disregarding the reverse flow of residual gas into the air chamber, in the case of negative-pressure scavenging, we have (see eq. (2), L_z with subscript 0):

$$L_{z0}/V_z = \left(1 - e^{-\frac{1{\cdot}336 \times 1{\cdot}08 \times 294}{1{\cdot}2 \times 290}}\right)\frac{1{\cdot}2 \times 290}{294} = 0{\cdot}829.$$

In this expression $T_e = T_s\,[2p_a/(p_s + p_a)^r] = 294° \text{ K}$, $r = (K - 1)/K$, and

$$V_z/V_h = 1 - 0{\cdot}5\,\sigma_e + \delta - 0{\cdot}925 = \frac{1}{1{\cdot}08}$$

Equation (5) then gives: $\lambda_{s0} = 0{\cdot}896$.

Taking account of the residual gases in the scavenge air passages, we obtain from eq. (24) (L_z with subscript 0'):

$$L_{z0}'/V_z = 0{\cdot}730,$$

the temperature of the gas-and-air mixture being

$$T_e' = 344° \text{ K}, \text{ so that } \lambda_{s0}' = 0{\cdot}860.$$

The residual gases also affect the scavenge ratio. Assuming that this effect is equally important in the case of reverse-flow scavenging, then, according to eq. (1), for an air consumption factor

$$S_z = SV_h\,p_a\,T_e/(V_z\,p_z\,T_a) = 1{\cdot}200,$$

related to the state of the gases within the cylinder, the scavenge ratio of the model is $\lambda_{sM} = 0{\cdot}80$, and the true scavenge ratio, according to eq. (5), is

$$\lambda_s = 0{\cdot}90.$$

Corrected for the effect of reverse flow, the scavenge ratio becomes

$$\lambda_s' = \lambda_s\,\lambda_{s0}'/\lambda_{s0} = 0{\cdot}865.$$

In the system considered, it will, however, be assumed that the scavenge air passages are so designed that no mixture of the residual gases of the reverse-flow occurs with the scavenge medium in the air chamber. Consequently, we can continue our calculations using $\lambda_s = 0{\cdot}90$.

Power Absorbed by the Blower: From eq. (10), assuming $\eta_{ad} = 0{\cdot}70$, we have: $H_{ad} = 2970$, and the blower power expressed as a mean pressure is then $p_g = 0{\cdot}670$.

Engine Output: As the various parameters are calculated for a given σ_a but for various σ_e values, while p_{za} and p_s are always kept constant, the total charge remains the same for all cases considered.

Thus, according to eq. (8), assuming $\lambda_g = 0{\cdot}765$, the mean effective pressure is: $p_e = 4{\cdot}53$.

The values of the parameters for different inlet slot heights are given in the following table:—

σ_e	S	ΔS	λ_s	p_e	p_g
0·28	1·657	0·143	0·912	4·43	0·83
0·26	1·565	0·105	0·905	4·46	0·78
0·24	1·336	0·089	0·900	4·53	0·67

Thus, in the example considered, the mean effective brake pressure is only very slightly affected by a change in inlet slot height. To obtain optimum slot dimensions, similar calculations are also required for various exhaust slot heights. Furthermore, the influence of the air consumption on cylinder wall temperature and piston temperature should be considered when making the final choice of the values for S, σ_e and σ_a.

C. MEASUREMENT OF THE CHARACTERISTIC PARAMETERS

To determine the optimum values for the principal dimensions of a scavenge system, the scavenge-ratio curve and the discharge coefficient must be known. The required data are obtainable from a model. However, if the tests are carried out on an engine, the suggested procedure is to measure the scavenge ratio, the air consumption factor, and the air supply ratio. These are useful in determining the causes of a possible deficiency in power output and in assessing the advantages of subsequent alterations of the scavenge system.

Of the various parameters required, only the air consumption factor and the scavenge ratio can be determined accurately with comparatively simple means. For the air supply ratio and the total charge, extensive equipment is required for accurate measurements.

1. Measurement of the Scavenge Ratio.

(a) *Engine Tests*: The scavenge ratio λ_s can be determined from the CO_2 content of the charge before and after the gas-exchange process. Then

$$\lambda_s = 1 - \frac{1}{\frac{u}{u'} - \xi\left(\frac{u}{u'} - 1\right)}$$

where ξ = water vapour content of the residual gases, and u, u' = CO_2 content before and after scavenging, respectively.

As the composition of the charge varies at different positions in the cylinder, the CO_2 content can only be measured adequately from gas samples representing the average composition of the charge. Therefore, the best method is to withdraw a large proportion of the charge, for instance, one third of the latter, from the cylinder during a definite time interval of the working cycle.

Provision for the arrangement of an additional valve for sampling can be made in the design of a test engine. On an engine already built, however, this would require considerable changes in the combustion chamber, and in such cases a much smaller auxiliary valve can be installed, and the measurement accuracy is correspondingly reduced. The valve dimensions depend primarily on the time interval available for sampling before gas renewal, that is, on the short time interval after completion of combustion up to the beginning of the scavenging process. For the sampling after the gas-exchange process, a considerably greater period of time is available. The two gas samples can be taken through the same valve, provided the valve control gear is adjusted to the particular timing of the engine.

(b) *Model Tests*: The scavenge model can be designed with either a fixed or a moving piston. Three different types will now be described.

In the first type, with a fixed piston*, the scavenge air is controlled by a valve situated immediately in front of the scavenge slots, the valve being opened by a falling weight and closed by a spring. Apart from the energy losses, the motion is equivalent to that of a piston with an infinitely long connecting rod.

*H. List, *Scavange Tests on a Small Two-stroke Cycle Cylinder Model*. Tungchi University, Woosung, China, No. 6, 1932.

Before operating the valve, the cylinder is filled with a mixture, the percentage of CO_2 and air of which is known, and the CO_2 content of the charge is then measured after the scavenging process. Although flow conditions are not exactly similar to those in the engine, comparative tests have shown that models of this type give sufficiently accurate results.

For higher speeds the scavenging process can be investigated by means of the model shown in Fig. 10. The piston is provided with an inflatable rubber cover, which is filled with compressed air so as to press against the cylinder wall. When the pressure is suddenly released, the piston is no longer retained and is pulled down by a weight until it impinges upon the piston rod support after having been accelerated through a given distance. The compressed air escapes through lateral slots in the cylinder, near the bottom dead centre. The motion is then damped by the elastic reaction force of compressed air acting on a braking piston fixed to the piston rod. The scavenge ratio is measured in the same way as for the slow-speed model.

For piston speeds corresponding to engine speeds above 1100 rpm, a scavenge model with a positively-controlled piston is indispensable. A model of this type has been developed according to the suggestions of A. Pischinger. Both this model and that of Fig. 10 enable a study to be made of the influence of cylinder pressure at inlet opening. Any required value for cylinder pressure can be obtained by adjusting the model before actuating the piston. This investigation would hardly be possible with a continuously-moving piston, owing to the excessive compressed-air consumption required.

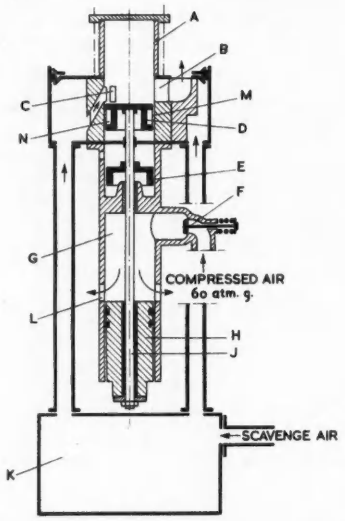


Fig. 10. **Scavenge model with pneumatically controlled piston.**

A glass cylinder, *B* outlet passage, *C* inlet passage, *D* working piston, *E* brake piston, *F* inlet for compressed air, *G* compressed air cylinder, *H* dropping weight, *J* piston rod, H scavenge air chamber, *L* exhaust slots for compressed air, *M* inflatable rubber cover over the piston, *N* compressed air space for pressing the rubber cover against the cylinder wall.

The piston is moved up and down once only during each test, by means of a continuously rotating cam, the piston stroke being somewhat greater than the maximum slot height. In the stationary condition, the piston is held in its upper position by holding members, which are pulled back for starting, and brought forward automatically again after completion of the stroke.

2. Measurement of the Air Supply Ratio.

The air supply ratio can be determined on the running engine either by measuring the excess-air ratio, or, if the quantity of residual gas is known, by determining the average CO_2 content of the residual gas.

When the first method is used, the air supply ratio is given by

$$\lambda_l = L_0 B [1 + (\lambda_v - 1) \lambda_s]/V_h,$$

while, with the second method,

$$\lambda_l = [L_a - V_a (1 - (u''/u))]/V_h,$$

where B = fuel consumption [kg/cycle], V_a = volume of residual gas [m³/cycle] referred to external conditions, u = CO_2 content of charge before scavenging, and u'' = CO_2 content of the residual gases.

The first method gives useful results only if the excess-air ratio is measured very accurately. Even when the second method is used, various difficulties have to be overcome, since the gas samples must be taken from the residual gas flow at certain instants during the cycle. The gas-sampling valve used for the second method must be located immediately after the exhaust slots. At a given adjustable instant during the scavenging process, it diverts a part of the exhaust flow during a certain number of revolutions. The gas samples thus taken from this gas flow have a constant composition. The CO_2 content u'' is the mean value for the gas-exchange process during a given number of revolutions.

3. Measurement of the Total Charge.

In many cases it is more convenient to make a direct measurement of the total charge, on the running engine, instead of first determining the air supply ratio.

For this, two different methods have so far been developed†. In the first method, the greater portion of the charge is ejected from the cylinder, during the compression stroke, into a measuring chamber through a sampling valve which can be switched on for a single cycle. The pressure rise in the auxiliary chamber is then a measure of the increase of its gas content. The gas still remaining in the cylinder is determined by a pressure and temperature measurement at the top dead centre.

In the second method, the temperature measurement can be omitted by employing a two-piece piston, the upper portion of which impinges upon the cylinder head and remains there after opening the sampling valve, so that the charge situated in the compression space is also driven into the measuring chamber.

4. Determination of the Scavenge Flow.

The study of the flow pattern enables a better appreciation of the relationship between the slot arrangement and the scavenge ratio. It also gives the possibility of determining correctly and in a short period of time the alterations which may be required. The flow within the cylinder can be followed either in a scavenge machine with a continuously moving piston, the flow being made visible by appropriate means, or in a scavenge model having a fixed piston, by measuring the flow velocities at different points.

The scavenge machine for the visual study of the flow according to the first method is made similar in design to the engine which is to be investigated. The portion of the cylinder above the slots, including the cylinder head, is made of glass. The piston slides in a cast-iron cylinder below the slots. The working cycle consists of compression, expansion, and scavenging. In order to simulate the pressure conditions of the

†E. Niedermayer, '*Scavenging Process in Two-stroke cycle Diesel Engines*. Forsch. Ing. Wes., Vol. 7, 1936, p. 227.
H. List and E. Niedermayer, *Tests on Two-stroke Diesel Engines*, Forsch. Ing. Wes., Vol. 8, 1937.

engine during the preliminary exhaust, compressed air is let into the cylinder through a special valve, during the downward motion of the piston. The flow is made visible by particles in suspension, which are intermittently illuminated by a stroboscope with constant-speed control. Tests with this equipment are being continued.

In the stationary scavenge model, a picture of the velocity distribution is obtained by determining the magnitude and direction of the air velocities. As important influences on the time variation of the process, such as, for instance, the piston motion, are lacking, the flow distribution of the engine cylinder is only approximately simulated. Furthermore, vortices are frequently generated, and the scavenge flow then continuously supplies energy to these vortices. In spite of these deficiencies, in many instances the results obtained on the stationary model can be utilised for an approximate assessment of the qualities of the scavenge system under investigation.

As the model is so designed that the measuring instruments can only be set up parallel to the axis of the cylinder, different types of pick-ups are required for the measurement of the horizontal and vertical components. The vertical components of flow through

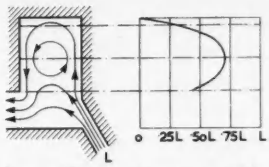


Fig. 11. **Stationary flow in scavenge model, and diagram showing the quantity of scavenge air flowing through the various cross-sections of the cylinder.**

a cross-section at a given level are plotted on a diagram for each level investigated, and the corresponding quantities of air flowing through are obtained by determining the volume occupied by the velocity crest of the cylinder section considered. Excepting scavenging of the uniflow type, for each horizontal cylinder cross-section the quantity flowing upwards must be equal to that of the downward flow. If, therefore, the air flow in the upward direction is found to be greater, this indicates the presence of a vortex passing through this section. Fig. 11 shows the proportion of the total quantity of air flowing through the model which passes through any given cross-section of the cylinder.

CZECHOSLOVAKIA

Nomogram for the Calculation of Condenser Bushings

By F. ČEŘOVSKÝ. (From *Elektrotechnický Obzor*, Vol. 36, No. 16, August, 1947, pp. 298-304, 6 illustrations.)

IN a detailed paper* Humburg has proved, that for the optimum utilization of materials in condenser bushings the axial stress should be constant. This view was confirmed by Einhorn several years later. Imhof suggested a method of calculation of the number of metal inserts by using a simple logarithmic scale. This method is based on the condition that the individual sections have equal voltage drops and capacitances. As the axial stress is to be constant, the axial steps of the individual layers will be uniform, but the thicknesses of the individual layers will not be equal. The thinnest and most stressed layer will be the one nearest to the terminal or the one nearest to the sleeve of the bushing. The calculation is carried out by choosing a suitable mean value of the radial stress and calculation of the corresponding maximum stress in the thinnest layer. If the maximum stress obtained is too high, the calculation has to be repeated with new values until the results obtained are satisfactory.

Polášek† calculates the external dimensions of the condenser bushing from the graphical solution of the transcendental equation and derives the lengths and thicknesses of the individual layers from the condition that the voltage across the individual layers have to be equal. Constant axial stresses and a given maximum radial stress are the basis of this calculation, and, therefore, it is not necessary to repeat calculations.

The method described in this paper is similar to the one developed by Polášek, and the author has placed the main emphasis on obtaining a definite decrease of the dielectric stresses and not merely a stress displacement. The conditions for obtaining reduced stresses are clearly defined in the nomogram for calculation of the main dimensions of condenser bushings which is given in this paper. The accuracy of the results obtained increases with increasing numbers of intermediary layers, and the ideal case is that where the number of intermediate layers is infinite.

*The Calculation of Condenser Bushings—*Archiv für Elektrotechnik*, 1923, p. 526).

†*Elektrotechnický Obzor*, 1936, p. 99.

Radial and Axial Stresses

Fig. 1 shows an axial cross-section of a condenser bushing. Usually the internal (terminal) radius r_0, and the length of the earthed outer layer, l_n, are given. The total length of the bushing l_0 and the radius of the outer layer, r_n, are functions of the test voltage E, and the permissible axial and radial stresses, λ_d and ρ_d. The terminal is the first electrode of the condenser series with the dimensions r_0 and l_0, and the outer layer (r_n, l_n) is the last electrode of the condenser series; any intermediate layer has the dimensions r, l. For an infinite number of layers, the elastance in any of the layers is given by:

$$d\left(\frac{1}{C}\right)=\frac{4\pi\,dr}{\epsilon\,2\pi\,r\,l}=\frac{2\,dr}{\epsilon\,r\,l}, \quad .. \quad (1)$$

where ϵ is the dielectric constant of the material. The

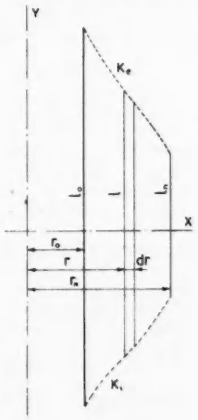


Fig. 1

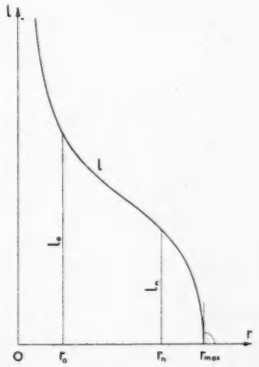


Fig. 2

elastance of the whole bushing will be the total elastance of the individual layers:

$$\frac{1}{C_0} = \frac{2}{\epsilon} \int_{r_0}^{r_n} \frac{dr}{r\,l} \quad .. \quad .. \quad (1a)$$

The potential difference de of any of the layers is proportional to the elastance of that layer and, as the terminal test voltage E is known, the potential difference of any layer can be calculated; since an infinite number of thin layers is assumed, the voltage at any point of the bushing is given by the relation:

$$\frac{de}{\int_E^0 de} = \frac{d\left(\frac{1}{C}\right)}{\frac{2}{\epsilon}\int_{r_0}^{r_n} \frac{dr}{r\,l}} \quad .. \quad .. \quad (2)$$

The limits of the first integral follow from the boundary conditions $e = E$ (terminal voltage) for $r = r_0$, and $e = 0$ for $r = r_n$, (earthed outer layer). Inserting

$$\int_E^0 de = -E,$$

and also values from eqs. (1) and (1a) into eq. (2), we obtain:

$$-\frac{de}{dr} = \frac{E}{r\,l \int_{r_0}^{r_n} \frac{dr}{r\,l}} = \rho, \quad .. \quad .. \quad (3)$$

where ρ is the radial stress in a given point.

To enable easier investigation of non-symmetrical condenser bushings a second reference axis X, perpendicular to the Y-axis, is introduced, and the only limitation to the position of the X-axis is that it should intersect l_n. If the bushing is imagined to be in a transformer cover, the upper part of the bushing will be in air and the lower part in oil and the X-axis can be imagined as the dividing line between the lower (oil-immersed) part l_i, and the upper part (air) l_e of the total length l. If the equation of the curves giving the lengths of the layers $K_e = f_e(r, l_e)$, and $K_i = f_i(r, l_i)$, is known, the axial stresses can be determined:

$$\left.\begin{aligned} \lambda_e &= \frac{de}{dl_e} = -\frac{de}{dr} \times -\frac{dr}{dl_e} = -\rho\frac{dr}{dl_e} \\ \lambda_i &= \frac{de}{dl_i} = -\frac{de}{dr} \times -\frac{dr}{dl_i} = -\rho\frac{dr}{dl_i} \end{aligned}\right\} \quad (4)$$

The two values λ_e, λ_i, can be replaced by a single value λ.

It follows from eq. (4) that:

$$dl_e = -\rho\,dr/\lambda_e, \quad dl_i = -\rho\,dr/\lambda_i,$$

and, as $l = l_e + l_i$,

$$dl = dl_e + dl_i = -\rho\,dr\left(\frac{1}{\lambda_e} + \frac{1}{\lambda_i}\right) = -\rho\,dr\frac{1}{\gamma},$$

where $$1/\lambda = 1/\lambda_e + 1/\lambda_i \quad .. \quad .. \quad (5)$$

and the two equations (4) are replaced by a single equation:

$$\lambda = -\rho\,dr/dl \quad .. \quad .. \quad .. \quad (4a)$$

but the relation $K = f(r, l)$ has to be known.

Equations (3) and (4) show that uniform radial and axial stresses cannot be obtained by suitable spacing of the metal foils, because for constant ρ and λ values we would obtain, according to eq. (4a):

$$dr/dl = -\lambda/\rho = \text{const.} \quad .. \quad (4b)$$

and the limiting curve of the individual layers would have to be a straight line; but from eq. (3) we obtain:

$$r\,l = \text{const.} \quad .. \quad .. \quad .. \quad (3a)$$

which is the equation of a hyperbola. It is obvious from eqs. (4b) and (3a), and (4a) and (3) that it is impossible to obtain uniform stress distribution in both axial and radial directions, and it is necessary to accept non-uniform stress distribution in one or both directions. Non-uniform axial stress, with uniform radial stress is useless in practice, because such bushings would be too long. Bushings with constant λ and variable ρ, and bushings where both λ and ρ are non-uniform have the same radial dimensions, but the former is shorter than the latter. Humburg has proved that a condenser bushing with a constant λ is the optimum solution.

Determination of the Limiting Curve of Condenser Bushings with a Constant Axial Stress (λ = Const.)

The equation of the limiting curve of the individual layers is obtained by inserting λ = const. in eq. (4a):

$$\frac{1}{r\,l}\frac{dr}{dl} = K_1 \quad .. \quad .. \quad (4c)$$

where K_1 is a constant; by integration of this equation we obtain

$$\log_e r = \tfrac{1}{2} K_1 l^2 + K_2, \quad .. \quad (6)$$

where K_2 is the integration constant. By introducing r_0, l_0 and r_n, l_n in the above equation we obtain the constants K_1 and K_2, and also the equation

$$\frac{\log_e r/r_0}{\log_e r_n/r_0} = \frac{l_0^2 - l^2}{l_0^2 - l_n^2}, \quad .. \quad .. \quad (6a)$$

which is the equation of the limiting curve $K = f(r, l)$.

For $r = 0$, $\log_e r = -\infty$, and therefore $-\infty = l_0^2 - l^2$, and $l = \infty$.

For $l = 0$,

$$\log_e \frac{r}{r_0} = \frac{l_0^2}{l_0^2 - l_n^2} \log_e \frac{r_n}{r_0}.$$

As l_0 is always larger than l_n, and r_0, r_n are finite positive values different from zero, r will also be a finite positive value.

$$\frac{dl}{dr} = -\frac{l_0^2 - l_n^2}{2\,r\,l \log_e \frac{r_n}{r_0}}, \quad .. \quad .. \quad (7)$$

is always negative. For $l = 0$, $r = r_{max}$ is a finite value, and

$$\left(\frac{dl}{dr}\right)_{r=0} = -\infty$$

For $r = 0$, $r\,l = 0$, and also

$$\left(\frac{dl}{dr}\right)_{r=0} = -\infty$$

and the limit curve of the bushing layers has the shape shown in Fig. 2.

Furthermore, let us determine the course of the curve

$$y = r\,l = r\sqrt{l_0^2 - \frac{l_0^2 - l_n^2}{\log_e r_n/r_0} \log_e \frac{r}{r_0}} \quad .. \quad (8)$$

For $l = 0$, $r = r_{max}$ is a finite value, and $y = 0$. For $r = 0$

$$y_{r=0} = \lim_{r=0} \sqrt{r^2 l_0^2 - \frac{l_0^2 - l_n^2}{\log_e \frac{r_n}{r_0}} r^2 \log_e \frac{r}{r_0}}$$

$$= \lim_{r=0} \sqrt{-\frac{l_0^2 - l_n^2}{\log_e \frac{r_n}{r_0}} r^2 \log_e \frac{r}{r_0}}$$

As $\lim_{r=0} r^2 l_0^2 = 0$,

$$\lim_{r=0} r^2 \log_e \frac{r}{r_0} = \lim_{r=0} \frac{\log_e r}{\frac{1}{r^2}} = \lim_{r=0} \frac{\frac{1}{r}}{-\frac{2}{r^3}} = \lim_{r=0} -\frac{r^2}{2} = 0$$

and $y_{r=0} = \lim_{r=0} r l = 0$.

The characteristic of the curve between the values $r = 0$, and $r = r_n$ is obtained from the derivation

$$y' = l - \frac{1}{2l} \cdot \frac{l_0^2 - l_n^2}{\log_e \frac{r_n}{r_0}} \quad .. \quad .. \quad (9)$$

For $r = 0$, $y'_{r=0} = \infty$;

for $r = r_{max}$, *i.e.*, for $l = 0$

$$y'_{r=r_{max}} = -\infty$$

The only extreme value occurs for

$$l_{extr} = \sqrt{\frac{1}{2} \frac{l_0^2 - l_n^2}{\log_e \frac{r_n}{r_0}}} \quad .. \quad (9a)$$

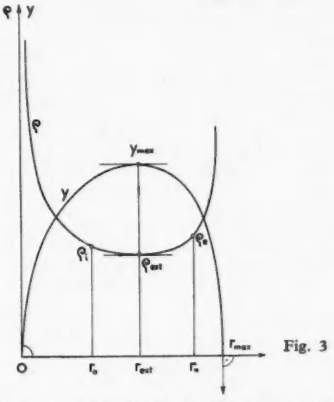


Fig. 3

This extreme value will obviously be a maximum, because r, l are positive values. The curve $y = r\,l$ will, therefore, have the shape shown in Fig. 3. The angle between the positive r-axis and the tangent is $+ 90°$ at the point $r = 0$, and $- 90°$ at the point $r = r_{max}$. The value y_{max} can be situated anywhere between $r = 0$ and $r = r_{max}$, depending on the choice of the constants, but a favourable condition is that $r_0 < r_{extr} < r_n$.

(*To be continued.*)

U. S. A.

Apparatus for Tensile Testing at Sub-Zero Temperatures

By E. J. RIPLING and G. TUER. (From *Product Engineering*, Vol. 20, No. 1, January, 1949, pp. 103-105, 6 illustrations.)

BECAUSE of the great importance of the mechanical properties of metals at low temperatures, an apparatus for conducting low temperature tensile tests has recently been designed and built at Case Institute of Technology. To keep the equipment comparatively simple, it was limited to tensile testing because this is the most general method of evaluating the strength properties of a metal, and is also generally considered the most lucrative mechanical test.

In designing the equipment care was taken to eliminate all sources of scatter which have their origin in the testing equipment and technique, thus reducing the number of tests necessary in each series to a minimum. Since most structural metals become brittle at sufficiently low temperatures, the mechanical properties under these conditions, as measured by a tensile test, are greatly affected by sight deviations from concentric loading. This is particularly true in determining the fracturing characteristics of these brittle materials since even slight eccentricity (or noncoincidence of the loading and specimen axes), if uncontrolled, produces a large amount of scattering.

To obtain maximum concentricity, the apparatus was designed to be used with a specially-built concentric tension testing fixture.* Earlier tests indicated that this equipment produced an eccentricity of loading of less than 0·001 in.

The method of loading used in the low temperature tests is shown in Fig. 1. The load is transmitted through all the parts by buttonhead ends which avoid eccentricities caused by threaded joints.

To allow for easy assembly and disassembly of these buttonhead-ended parts, it was necessary to design some of the members split. All these split members are held together by tapered slip rings. Buttonhead end loading members were ground to size on the cylindrical surface, to fit snugly with adjacent split members. Surfaces under the heads were ground to planes perpendicular to the cylindrical surfaces.

The main mass of the loading members are separated from the specimen by large insulating joints which allowed the specimen to be held at the temperature of liquid nitrogen while the loading members furthest from the joints are still not much lower than room temperature.

At "intermediately" low temperatures (—20 to —40° C), load was transmitted through the insulating joints by means of steel washers, since the length of the loading members was found to be sufficient insulation. At lower temperatures, the steel washers were replaced

*See G. Sachs, J. D. Lubahn, and L. J. Ebert, Notched Bar Tensile Test Characteristics of Heat Treated Low Alloy Steel," *Trans.*, ASM, Vol. 33, 1944, pp. 340-395.

with thermosetting plastic laminate washers. However, because of the greatly reduced impact strength of the plastic washers at low temperatures, it was necessary to place thin sheets of neoprene in the insulating joints between the loading members. This absorbs the impact of the fracturing specimen, and prevents cracking of the plastic washers.

To change specimens during testing, only the parts between D and F (Fig. 1) must be removed from the testing set-up. When these parts are disassembled, they are heated to room temperature and dried before re-assembly. This prevents the formation of ice which might result in eccentric loading.

The eccentricity of loading of this complete assembly was determined by obtaining stress-strain curves in the elastic region on opposite sides of a test specimen by means of two Huggenberger lever-type strain gauges.

Since this equipment was to be used to obtain point by point stress-strain curves at various low temperatures, a method for holding the specimen at one temperature for a considerable time was developed. An insulated flask was made to surround the specimen and the loading members adjacent to the specimen. The coolant, usually isopentane, with which the flask was filled, was held at the desired temperature by passing liquid nitrogen at a definite rate through a copper coil on the bottom of the flask.

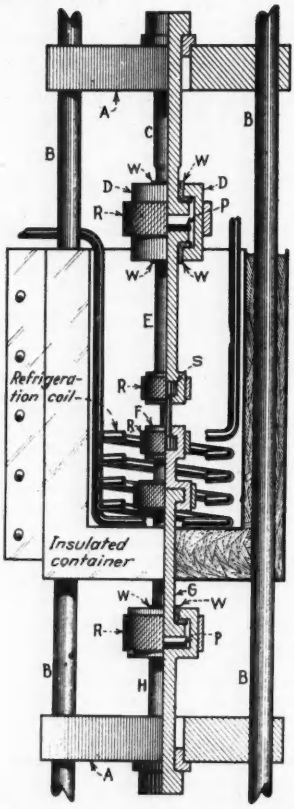


Fig. 1. **Loading members, refrigeration coil and insulated flask, in half-section.**

A = Concentric loading head; B = Guide rods; C, G = Solid loading members; D, E, F, H = Split loading members; P = Neoprene shock absorbers; R = Slip rings for split loading members; S = Specimen; W = Split plastic insulating washers.

The equipment used to control the isopentane temperature is shown in Fig. 2. Since the nitrogen in the container is constantly absorbing heat, it evaporates continuously, to build up a considerable pressure. This pressure controls the rate of flow of the liquid nitrogen. As shown in Fig. 2, two escapes were built into the nitrogen container, one for gas and one for liquid nitrogen. The escape channel for the gaseous nitrogen has a built-in stop-cock to control the rate of flow of escaping gas. Since pressure built up by the vaporizing nitrogen is not always large enough to develop the desired rate of nitrogen flow, a pressure bulb was added.

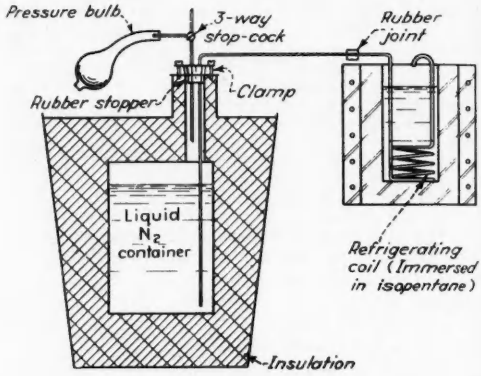


Fig. 2. **Method of controlling temperature of coolant (isopentane) in flask. Liquid nitrogen passes through refrigerating coil.**

When the specimen was to be tested at the lowest temperature, the refrigerating coils were not used. Instead, the liquid nitrogen was forced directly into the cooling flask surrounding the specimen at a rate which kept a constant level of nitrogen in the container.

To change specimens, it was necessary to design the cooling chamber with a removable front. This presented a problem of finding suitable gasket materials. It was found that neoprene sheeting was satisfactory with isopentane, and a heavy grease was satisfactory with liquid nitrogen.

At temperatures between —80° C and room temperature, the refrigerating coil caused too large a temperature fluctuation during the test, so that equilibrium mixtures were used. A convenient mixture was dry ice in a pentane of gasoline for—78° C.

When liquid nitrogen was used as the refrigerant with isopentane, the temperature fluctuated about ± 2° C; also, the temperature difference between the top and bottom of the specimen was generally found to be about 2° C. When equilibrium mixtures were used (liquid nitrogen, dry ice and pentane), these fluctuations were smaller than the sensitivity of the temperature measuring equipment.

To obtain strain data, it was necessary to develop a remote reading strain gauge (mechanical lever-type) because readings could not be made directly on the specimen.

Initial and final diameters of all specimens tested were measured on a microprojector. These values were used in calculating the strength and ductility properties of the material.

Temperatures were measured by placing the " hot " junction of two thermocouples in the two centering holes of the test specimens. These thermocouples, which represented temperatures at the top and bottom of the specimens were then run through a " cold " (0° C, ice water) junction to a portable potentiometer.

WORKSHOP TECHNOLOGY

Internal Resonance—One of the Principal Causes of Machining Vibrations

By L. B. EHRLICH. (From *Stanki i Instrument, Russia*, No. 1, 1949, pp. 20-22, 2 illustrations.)

INTERNAL resonance has long been known to the practical engineer as one of the causes of vibration on all kinds of machine tools, but no investigation into its nature has so far been carried out. It is a well-known fact that it is impossible to avoid vibration when carrying out a finishing cut with a bracket type internal boring tool holder, if the ratio of length to diameter of the machined part exceeds 4 to 1.

Developments in hard alloys permit of adopting a length ratio of 7·3 to 1 in place of 4 to 1 for steel. Internal grinding machines are equipped with a large number of holders and wheels for different ratios of diameter to length of hole, but none of these permits a ratio exceeding 5 to 1.

When turning long objects, the general rule is to use a steady rest when the ratio exceeds 10-12 to 1.

The above facts show that the vibration occurs on all kinds of machine tools for most forms of machining operations and that the vibration starts when a certain length-to-diameter ratio is exceeded for the machined part or tool holder, and depends on the shape, mechanical properties and methods of fixing.

Internal Resonance: Vibrations of rods can be longitudinal, transverse or torsional, and while, theoretically, they are independent of each other, if it is permissible to neglect the squares of deformations, in practice calculations used to be carried out in each case solely for the predominant form of vibration, with the knowledge that each one of the three forms of vibration would occur to a certain degree. While

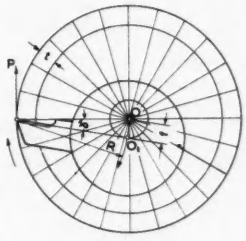


Fig. 1

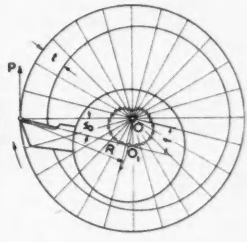


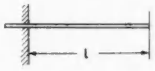
Fig. 2

the transverse vibration of a rod may be unimportant in itself and of a lower frequency than the other two, it is possible for a higher harmonic of it to be in resonance with, say, the longitudinal frequency. In a case of this nature, small irregularities may have important effects, and this type of resonance has been given the name of "internal resonance."

Internal Resonance of a Boring Tool Holder: Figs. 1 and 2 show the position of a boring tool holder for concentric and eccentric positions. During the operation the centre of the holder moves from O to O_1, the distance being proportioned to the amount of cut, and the cutting angle becomes equal to r.

If a small disturbance occurs the sag F will change,

TABLE.

	Type of Fixing	Transverse Vibration	Torsional Vibration
(a)		$N=\left(4{\cdot}32\times10^6\times\frac{d}{l^2}\right)\times1;\ 6{\cdot}35;\ 17{\cdot}5\ldots$	
(b)		$N=\left(12{\cdot}1\times10^6\times\frac{d}{l^2}\right)\times1;\ 4;\ 9\ldots$	$N=\left(4{\cdot}85\times\frac{10^6}{l}\right)\times1;\ 3;\ 5\ldots$
(c)		$N=\left(19{\cdot}1\times10^6\times\frac{d}{l^2}\right)\times1;\ 3{\cdot}22;\ 6{\cdot}62\ldots$	

the new position being obtained after one oscillation. The prevailing forces may originate a torsional, bending or combined vibration. In the last case, the excitation of vibration is considerably facilitated and will occur even with minute excitation forces. In practice, a pure torsional or bending vibration will never occur, for even if the vibration starts as one of the pure forms, it will always lead ultimately to a combined vibration, and the question as to whether internal resonance will occur will merely depend on the respective frequencies. The detailed description given for a boring tool holder applies equally to other methods of machining.

Quantitative Relations: From the formulae in the table it will be seen that the frequency of transverse vibrations is always lower than that of torsional vibrations for rods with $l \gg d$.

The second harmonics of one vibration will coincide with the basic frequency of the other for the following length-to-diameter ratios (see also table):

(a) $\frac{l}{d} \approx 5{\cdot}7$; (b) $\frac{l}{d} \approx 10$; (c) $\frac{l}{d} \approx 12{\cdot}7$.

For metals other than steel, these figures must be multiplied by $\sqrt{G/q}$ for torsion and $\sqrt{E/q}$ for bending, where q is the unit weight of the rod.

For alloyed steel, the above figures would, therefore, have to be multiplied by about 1·26, giving 7·2 in case (a).

Whereas, in theory, the ratios calculated are critical values, in practice it becomes obvious that they are simultaneously maximum permissible figures, as, if they are exceeded, bending of the rod becomes excessive.

From the considerations in this article it is evident that properties which would raise both the frequency of torsional and bending vibrations do not help in reducing the resonance, and to achieve this it is necessary to widen the gap between the two frequencies, or, in other words, the ratio between E and G.

Wire Loop-Forming Tool

By BÖTHIG. (From *Werkstatt und Betrieb, Germany*, Vol. 81, No. 12, December, 1948, p. 367, 5 illustrations.)

THE wire loop shown in Fig. 1 can be produced by a single tool as shown in Fig. 2. The tool consists of a forming plunger *a* in plunger head *f*, which is spring-loaded, and two horizontal slides *b*. The compression spring in the plunger head must be strong enough to transmit the force required for bending the wire into U-shape. The tool head, which is guided on two vertical columns, also carries two wedge members *c* for actuating the side strokes of slides *b*. By pressing down the tool head, the forming plunger *a* bends the wire over the jointed forming lugs *e* hinged in slides *b*. The wire is now bent into U-shape in die *d*. During continued downward motion of the tool head, the wedge members *c* actuate the slides *b*, which, moving inwards, bend the wire ends outside die *d* over the forming plunger *a*. The latter is shaped to receive the bent-over wire ends, the hinged forming lugs (Fig. 3) enabling the inward and downward bending of the wire ends. Accordingly, the appropriate part of the forming plunger (Fig. 4) must be given an inclined surface to produce the required final shape of the wire loop.

The forming plunger should be prevented from rotating. This is done by providing the plunger head *f* with a feather key, which slides in a keyway cut in the disc *g*.

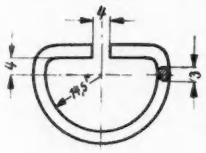


Fig. 1. **Wire loop (of 3 mm dia. wire).**

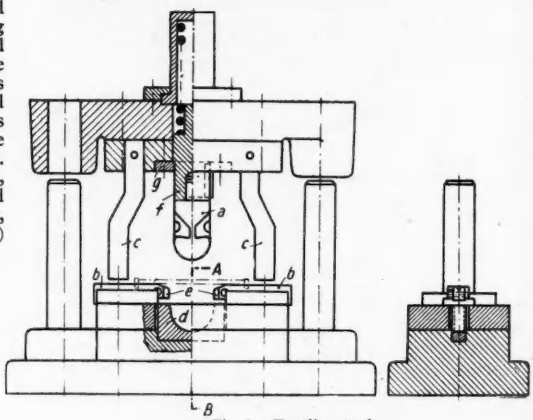


Fig. 2. **Bending tool.**
(*a*) forming plunger; (*b*) horizontal slide; (*c*) wedge member; (*d*) die; (*e*) hinged forming lugs; (*f*) plunger head; (*g*) disc.

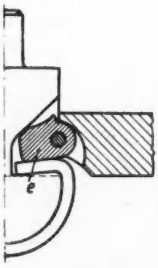


Fig. 3. **Jointed forming lugs *e* in completing forming process.**

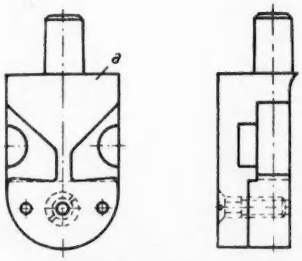


Fig. 4. **Forming plunger *a* shaped to receive inclined wire ends.**

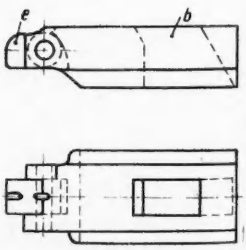


Fig. 5. **Horizontal slide *b* with jointed forming lug *e*.**

Barrel Burnishing of Light Alloy Products

By CH. ETIENNE. (From *Revue de L'Aluminium, France*, Vol. 25, No. 150, December, 1948, pp. 377-382, and Vol. 26, Nos. 151 and 152, January and February, 1949, pp. 15-21, and 65-69, 45 illustrations.)

(*Concluded from May issue.*)

(b) *Cleaning*

Oxide skins and surface impurities must be removed by chemical cleaning, i.e., by brief and slight etching. If the etching is too sharp, the result will be a grained or uneven surface which cannot be highly polished by burnishing. Pure Al-components and Al-Mg and Al-Cu alloys are etched for 30 to 90 seconds in a 5 per cent soda bath at 70° C. They are then vigorously rinsed under running cold water and neutralised in a nitric acid bath for the removal of any skin or staining deposits. Small articles treated in bulk should be stirred during neutralisation, and the nitric acid concentration may have to be raised to 20 per cent and the time of treatment to 5 minutes. Serious contamination with copper salts (Duralumin treatment) may impair the finish obtainable. After another rinsing operation, the articles are left in water or immediately burnished. They must not be allowed to dry. Al-Si alloys with more than 4 per cent Si should be immersed for 3 to 5 seconds in a cold bath containing 30 per cent by volume of hydrofluoric acid (50 per cent), 30 per cent by volume of nitric acid (36 deg. Be.), the rest water, and should afterwards be vigorously rinsed in cold water. If pieces are brazed, even this severe treatment will not remove the brazing stains.

F. DESIGN FOR BURNISHING

Small articles treated in bulk must be so designed that they cannot stack, hook, or bind together. Flat plates tend to swim on the bed of balls and are easily deformed unless slightly cambered. Recesses which retain the balls should be avoided; small holes may be drilled after burnishing. Delicate articles should be made of a hard alloy. If a high lustre is required, a suitable alloy must be chosen. The surface quality before burnishing influences the result. A buffing operation may be avoided if articles are chill cast instead of sand cast, or forged instead of cast, or if they are machined. Dies and tools should be treated with special care.

G. FAULTS AND REMEDIES

(a) *If articles, although clean, are not properly polished*:—

Check composition of lubricant, quality of soap. Excessive charge restricts ball movement. Too many articles in the charge. Lubricant used up or balls to be regenerated.

(b) *If articles come out uniformly grey or black*:—

Too many articles in the charge. Timber of drum unsuitable. Balls are oxidised or covered with Al-skin and should be regenerated. Lubricant and articles are dirty. Neutralisation is incomplete; the bath should be renewed or its concentration increased. Al-Si articles have not been rinsed or brushed vigorously enough. Composition of alloy is abnormal and requires special pre-treatment. Lubricant not prepared with distilled water. Soap too resinous or too strongly alkaline. Traces of soda left after regeneration of balls; rinse drum and balls in cold water.

(c) *If the articles are deformed*:—

Bed of balls too high, take smaller barrel. Bolt heads or other details project into interior of barrel. Charge of balls too small for delicate and large articles, which then foul each other. Total charge too small, balls fall from too great a height. Speed too high, balls bounce. Bad design of details.

(d) *If the articles are marked by impact of balls*:—

Speed too high or charge too small. Articles too heavy; increase amount and/or diameter of balls.

(e) *If the articles show a uniform-grained skin*:—

Amount of balls too large for articles of a soft alloy. Ball diameter too small. Material is coarse-grained due to faulty rolling or over-etching.

(f) *If the articles are scratched*:—

Scratches existed before, and are accentuated after burnishing. Buffing with coarse-grained wheel. Amount of balls too small for articles with sharp projections. Re-design articles. Cracked balls or foreign particles in charge. Projecting bolt heads. Flat or hollow articles swim on bed of balls; increase amount of balls or mix balls of different diameters and shapes.

(g) *If the articles are stained*:—

Insufficient cleaning or neutralisation. Grease stains re-appear after burnishing. Pieces have dried after rinsing in calcareous water. Burnishing bath used up or has not been stirred for 10 minutes before start of treatment. Surface impurities have not been removed.

(h) *If white stains appear on the articles*:—

Over-etching. Reduce concentration of soda bath to 5 per cent for delicate articles. Buffing too intense. Coarse grain or rolling defects in material. Drying after rinsing in calcareous water. Burnishing liquid unsuitable or contaminated by soda.

(i) *If black scratches appear*:—

Burnishing bath used up; black deposits appear in scratches left from previous working. Inclusions in rolled material.

H. SPECIAL BURNISHING MACHINES

The movement of balls and articles in a drum is quite irregular and better results can be obtained in a shorter time if machines are used which give a regular motion by either forcing the pieces to move in an ordered fashion through a stationary bed of balls or by moving the balls over articles held stationary. In the

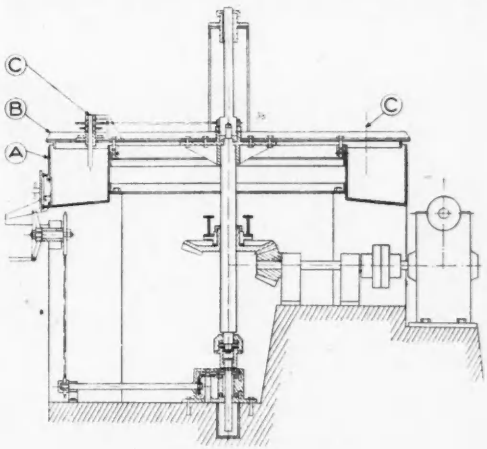


Fig. 9. Schematic drawing of burnishing machine with vertical axis.

A = annular container; B = rotating platform; C = rotating mandrel carrying the pieces. The hand wheel serves to adjust the height of platform B for loading and unloading.

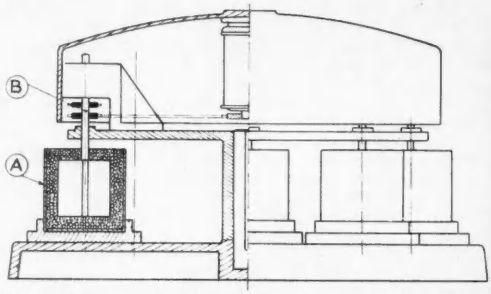


Fig. 10. **Schematic drawing of burnishing machine for hollow containers.**
A = container to be burnished ; B = spindle with blades.

machine Fig. 9 the articles are held on the rotating mandrels and describe a cycloidal motion. The platform B moves at about 200 fpm measured at the mandrel, but the mandrel rotation must be fixed more carefully according to platform speed, size and shape of articles, and ball diameter. This machine requires more power than a barrel, but the finish is superior, and the time of treatment is very much reduced, i.e., to about 2 minutes for hard alloys, and to 20 seconds for high purity Al. The danger of deformation is minimised since the articles can be suitably secured. The handling is very simple. In the machine Fig. 10, a fan wheel moves in a bed of balls inside the hollow container to be burnished. The machines give a good polish at very low cost. Articles which cannot be polished mechanically or which are too delicate for drum polishing can be treated. However, certain particular faults sometimes appear, e.g., annular or linear markings if the platform speed is too large relative to the speed of rotation of the mandrel, black traces due to seizure if the rotational speed is too high, and wear of sharp edges due to the greater burnishing pressure. This can be reduced by a decrease in rotational speed.

I. APPLICATIONS

Figs. 11 and 12 show some applications of the burnishing process. It is extensively used in the mechanical and electrical industries, for series production of office equipment, surgical instruments, jewellery, household articles, furniture fittings, hardware, metal buttons, needles, musical instruments, watch and clock manufacture, the toy industry, etc.

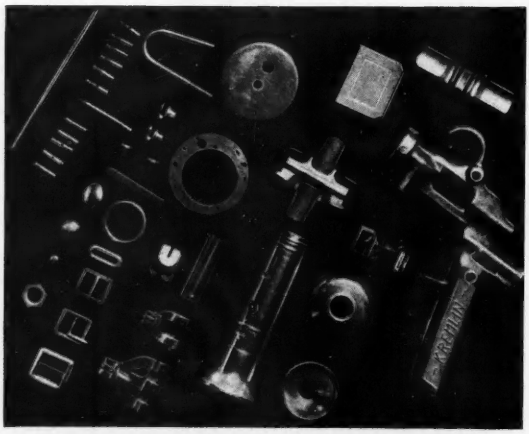


Fig. 11. **Examples of pieces cut from bar, die cast, worked, or punched, and ball burnished, for the electrical industry, hardware and sports articles.**

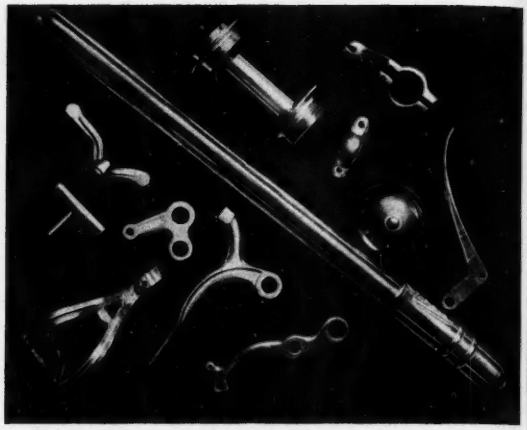


Fig. 12. **Several ball burnished articles used in the cycle industry.**

"Poke" Welding—A New Method of Joining Stainless Steel, Aluminium and Mild Steel

By F. J. PILIA. (From *Materials and Methods, U.S.A.*, Vol. 29, No. 3, March, 1949, pp. 64-67, 5 illustrations.)

A NEW spot welding process called "poke welding," making use of the inert gas shielded arc principle, has been developed to fill a gap that existed between resistance and fusion welding. It is being successfully used to join mild steel, stainless steel, aluminium, etc.

A spot weld is produced by heat from an electric arc between a tungsten electrode and one surface of two lapped pieces of metal. It differs from a resistance spot weld in that it is a result of fusion only, without pressure being applied to the electrodes. The welding action is controlled by the current input of the arc and the length of time the arc impinges on the surface of the material being welded. The electrode, the arc and the fluid puddle are shielded with an inert gas in the usual manner.

The tool for performing the weld consists of a gun of moulded plastic. The tungsten electrode is located in the centre of a water-cooled copper cup which bears against the workpiece. Argon gas is distributed smoothly through the interior of the cup by a series of orifices in the electrode holding mechanism. Three hoses and one switch cord are encased in a rubber jacket approximately 25 ft long to supply the welding current, cooling water, argon gas, and trigger control of the gun. So far, the process has been used successfully on mild steel and stainless steel up to 0·078 in. in thickness, and on some aluminium alloys.

Perhaps the major advantage of the process is its ability to produce welds from one side of the joint only, as no forging pressure is required. This makes poke welding especially suitable for jobs at places accessible only with difficulty on one side of the joint.

Furthermore, welds can be made without disturbing the reverse side of the sheet being welded.

Because the arc is primarily within the cup, only goggles or plastic face shields are required, no toxic fumes, smoke or spatter are produced and the operator is less inconvenienced than happens in the case of other welding processes. A series of examples concludes the article, showing the usefulness of the process in practice.

Self-Controlled Heating Element

(From *The New Zealand Electrical Journal*, Vol. 22, No. 1, January, 1949, p. 63, 1 illustration.)

IN this unit, the usual bimetallic strip, tube and rod, or liquid-bellows principles of heat controls have been dispensed with, as the casing of the element is shaped to allow it to expand and contract without deformation and can be used for one expansion ratio, whilst the centre stud is used for the other. The expansion differential of these two parts is multiplied by levers so that the contacts show a clean break for a temperature rise within one or two degrees. This type of control together with the shape of the element, which gives perfect surface contact with a separate container, reduces the lag in control of temperature to the minimum. The electrical efficiency of the element is also exceptionally high, nearly equivalent to that of an immersion heater.

As the heat given off by this element is directly controlled, the element offers many advantages, both from a technical standpoint and from the point of view of the user, whether it be a housewife or an industrial user of glue, gelatine, wax, etc. Fig. 1 shows the construction of such an element. It will be noted that expansion breaks the contacts, and spring pressure closes them, ensuring safety in operation. Liberal use has been made of stainless steel for the parts. An element of this type used in a household appliance is rated at 650 Watts, 230 Volts a.c. The control allows a temperature range of 120 to 420° F. Although the wattage is low, water boils faster than with a conventional hot plate. With temperature control, it has, of course, many advantages over the hot plate, particularly for slow simmering and for keeping foods warm. The various utensils available for the base element make it a complete cooker in itself, even for the baking of small cakes. A similar element is used for industrial applications, and can maintain temperatures within plus or minus 2·5° F.

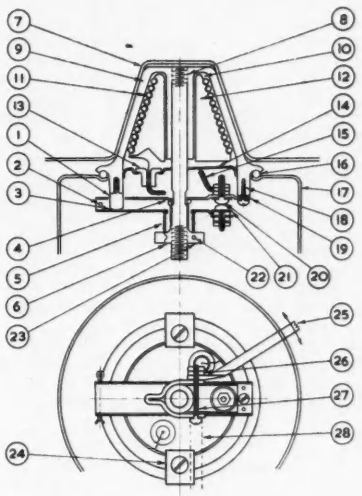


Fig. 1

(1) Thrust stud; (2) Hinging pin; (3) Fulcrum; (4) Fulcrum adjustment; (5) Closing spring; (6) Control nut; (7) Container; (8) Stud; (9) Casing; (10) Cement; (11) Element; (12) Former; (13) Bushes; (14) Cap; (15) Leads; (16) Asbestos; (17) Cover; (18) Holding screw; (19) Bottom lever; (20) Silver contacts; (21) Top lever; (22) Screw worm adjustment; (23) Control lever; (24) Fixing clamps; (25) Lever control; (26) Locking nuts; (27) Worm adjustment; (28) When screw fixed control an extended screw used in place of lever.

Belt Drives

By C. A. NORMAN. (From *Machine Design, U.S.A.*, Vol. 21, No. 2, February, 1949, pp. 111-115, 9 illustrations.)

WHAT is the belt tension which will ensure dependable power transmission and reasonable belt life, even under sustained heavy load? Need for an answer to this question, with particular reference to front-end belt drives on certain automotive vehicles, prompted the Goodyear-sponsored investigation discussed in this article. These drives involve high speeds, small contact angles, and also small pulley diameters. It was, therefore, necessary to determine the influence of contact angle on transmissive power with the small pulley diameters used.

Among the phases studied were the effects of centrifugal force, fixed and floating drives, angle of contact, ratio of pulley diameter to belt thickness, and ratio of total tension to initial tension.

It was found that centrifugal force may have much less influence than commonly assumed.

The basic formula for belt transmission has been

$$\frac{T_1 - \frac{wv^2}{g}}{T_2 - \frac{wv^2}{g}} = e^{fa/\sin\gamma} \quad \ldots \quad \ldots \quad (1)$$

where T_1 = tight side tension, lb; T_2 = slack side tension, lb; w = weight of belt, lb per ft of length; v = belt speed, fps; g = gravity acceleration, ft per sec per sec; e = basis of natural logarithms; α = contact angle, radians; f = coefficient of friction; and γ = half the groove apex angle (γ = 90 deg. for flat pulleys).

Equation (1) is derived on the assumption that when the belt runs over the pulley both tensions are reduced by the amount wv^2/g, and that there is nothing to compensate for this reduction. According to this formula, if wv^2/g equals the slack side tension T_2, power transmission ceases, since $T_1 - wv^2/g$ would then also have to become zero, $e^{fa/\sin\gamma}$ not being equal to infinity.

This, as was realized earlier in Europe and is now becoming realized in this country, is wrong in fundamental theory and amounts in practice to a mere approximation which is more and more wrong the slacker the belt can run, and the less it stretches elastically.

Formulas allowing for influence of centrifugal force must therefore contain terms reflecting the influence of the tension on sag and belt length. A complete formula* for this influence in the case of a horizontal, idling belt shows that the difference between the standstill shaft pull per side, T_0, and the running shaft pull, T_0, is equal to the old-time quantity wv^2/g, modified by three terms, the first allowing for a difference in

* See *High-Speed Belt Drives*, Engineering Experiment Station Bulletin No. 83, Ohio State University.

shaft distance, the second for elastic stretch in the belt, and third for belt stiffness.

Before proceeding to a discussion of the tests it should be remarked that centrifugal force, as such, does not bulge the belt and in fact gives the belt between the pulleys no definite shape whatever. If a belt bulges, this is due merely to stiffness and actual tests show that the bulging force is no greater at 7000 fpm than it is at standstill. All this is of course in harmony with the fact that neither the radius of curvature nor the included angle appears in the formula for the centrifugal tension wv^2/g.

In Fig. 1 is shown a plot of idling tests run to verify the stretching action of the centrifugal force. The driven pulley was mounted on a sliding carriage and the belt was a smooth cotton V-belt which is fairly stretchable elastically as modern belts go. The pull on the driven shaft was first held barely sufficient to keep the belt running straight. The increase in centre distance as the belt was speeded up to over 8000 fpm is plotted as abscissa. The belt was then idled slowly at 70 rpm and dead loads were applied at the driven end to produce a stretch equal to that produced by the previous centrifugal force. The practically perfect agreement between the computed centrifugal tension wv^2/g and the dead load per side for equal stretch should be noted.

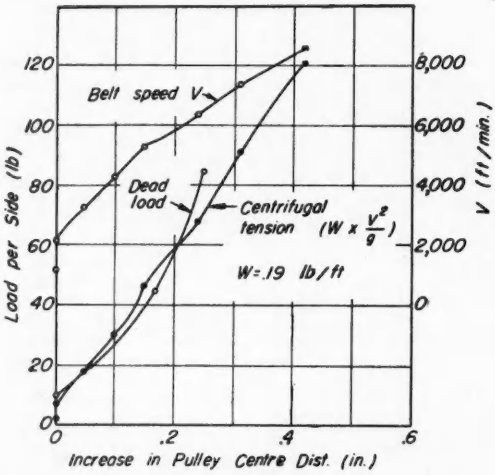


Fig. 1. **Results of centrifugal force tests with C-105 smooth cotton V-belt on 9-inch o.d. pulleys.**

The really important test results are typified by Figs. 2 and 3. Fig. 2 shows that it is possible to transmit more effective pull at high speed than at low speed. This applies up to speeds of over 6000 fpm and also was verified for two different lengths of steel cable V-belt at two different initial tensions. The slips, while they increase with the load, do not seem to increase with the speed for constant load.

The tests just discussed were carried out on horizontal drives with the driven shaft journaled in a sliding carriage and the torque taken through a flexible shaft to a Prony brake. The shaft distance was locked after the initial tension had been set by dead weights.

In the tests reported in Fig. 3 the driven shaft carried a calibrated fan. The pulleys were the same as before, but the belts were only 68 inches long. They were of steel-cable type as well as cotton-cord type. The main interest centres in showing the very much greater

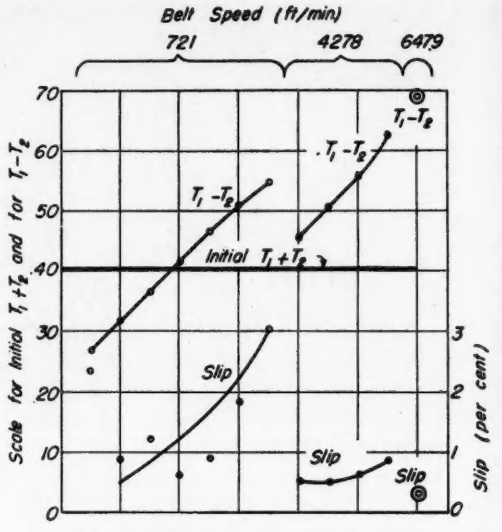


Fig. 2. **Comparative speed tests with C-96 smooth steel-cable V-belt on 9-inch o.d. pulleys, locked centre distance.**

slip of the cotton-cord belts. The slip was also greater for the stiffer smooth belts than for more flexible notch belts.

If the belt does not stretch, the intimacy of contact between the belt and the pulley is not changed by the centrifugal tension. The pressure between pulley and belt will be determined by any applied initial tension T_0. If the belt stretches, the tension T_0 will be influenced by the sag, and the belt pull against the pulleys will be determined by the sag condition. With vertical belts, stretch due to centrifugal tension may actually pull the belt free of the lower pulley.

FIXED AND FLOATING SHAFT DISTANCE: The new and somewhat surprising thing shown in Fig. 2 is that with a fixed shaft distance it is possible to transmit an effective pull $T_1 - T_2$ greater than the initially applied total tension $T_1 + T_2$. There is one possible explanation of this phenomenon which, however, will apply only to certain types of belts. For short or vertical belt drives Carl G. Barth suggested a formula for the relation between the initially applied tension per side T_0, and the working tension T_1 and T_2

$$\sqrt{T_1} + \sqrt{T_2} = 2\sqrt{T_0} \quad .. \quad .. \quad (2)$$

This formula, however, applied only to belts the stress-strain curve of which would run as curve A in Fig. 4, that is to say, the relation between belt length l and tension T could be represented by an equation $l = a + b\ T^m$, where a and b are constants, and m an exponent less than unity.

For such a relation, as Fig. 4 shows, an increase in tension from T_0 to T_1 on the tight side would correspond to a smaller stretch than for an equal amount of tension drop from T_0 to T_2 on the slack side. Hence, as load was applied, the belt would shorten. If the shaft distance were kept constant, the belt would tighten.

If the stress-strain relation is linear as represented by curve B, the extension on the tight side would equal the shortening on the slack side, and the belt length would remain constant. With such a relationship

$$T_1 + T_2 = 2\ T_0 \quad .. \quad .. \quad (3)$$

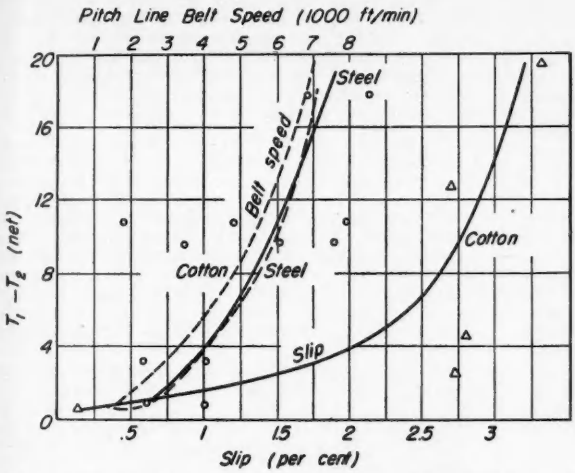


Fig. 3. **Comparative high-speed tests with C-68 notch belts on 9-inch o.d. pulleys, with drive locked at $T_1 + T_2 = 30$ lb.**

Finally, if the stress-strain curve bends towards the strain axis, as curve *C* in Fig. 4, the extension on the tight side would be greater than the contraction on the slack side. The belt length would increase with the load and

$$T_1^n + T_2^n = 2T_0^n \quad .. \quad .. \quad (4)$$

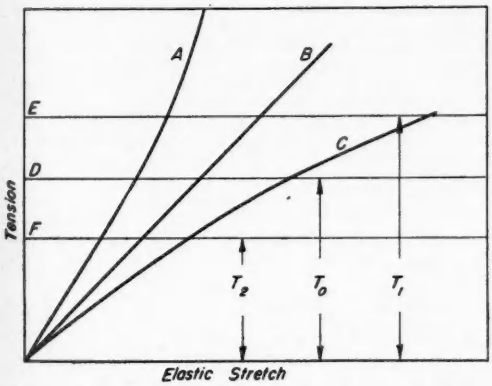


Fig. 4. **Typical strain curves for various types of belts.**

where n is greater than unity. Total tension under load would be smaller than the initial tension, and the effective pull transmissible with fixed shaft distance would be smaller than the load transmissible with a floating drive.

In stress-strain determinations at the Ohio State University it was found that the majority of V-belts had stress strain-curves of type *C*. In spite of this they transmitted more load with fixed shaft distance than with floating drive.

The question was cleared up by mounting the driven shaft on a sliding carrier, as in Fig. 5. The driven shaft either carried a fan, or was connected through a flexible shaft to a Prony brake.

It was found that when the tension on the slack side approaches zero, if excessive slip does not occur the drive becomes a hoist, the tight side pulling the driven pulley toward the driving pulley. This of course increases the slack and decreases the contact angle on the pulleys so that increased slip and, finally, breakdown of the drive occur. To keep the shaft distance constant, it is necessary to increase the load *L*. This increases the contact angle, as well as the tensions, and permits the transmission of a greater effective pull. Hence a fixed shaft distance is equivalent to adding to the total tension, and permits the transmission of a greater effective pull without excessive slip than does a floating drive.

A horizontal drive with a nominal contact angle of 180 degrees represents, of course, the most favourable condition for transmitting large effective pulls, $T_1 - T_2$, even with very low slack-side tensions. With smaller contact angles the higher transmissive power obtainable with fixed shaft distance might not be so noticeable. This advantage, therefore, had to be established by direct tests on such drives which showed that with the same initially applied tension, more effective pull could uniformly be transmitted with fixed shaft distance than with a floating tensioning device. The advantage of the floating drive with tensioning device lies in compensating for permanent belt stretch.

CONTACT ANGLE: The first test results to be plotted were for C-size belts on 6·9, 5·0 and 4·0-inch outside diameter pulleys. All these pulley diameters are below the absolute minimum of about 9-inch o.d. recommended for long-life industrial drives. Tests with C-size belts on 9-inch pulleys, however, showed hardly any influence of contact angle between angles of 90 and 180 degrees, but it should be noted that these tests were run with effective and total pulls considerably below the rated values for these belts. If any conclusion is to be drawn from this it would be that the results were due to the comparatively long arc length of contact on large pulleys as well as, of course, to the large radius of curvature and consequently lessened influence of belt stiffness in reducing contact angle.

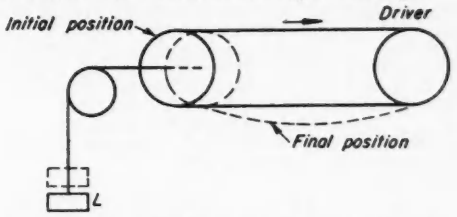


Fig. 5. **Floating belt drive, showing method of applying initial tension for the test set-up.**

Strange to say, a similar independence of contact angle may be observed also on very small pulleys. The reason then is that belt stiffness reduces the actual contact between belt and pulley almost to a point for all angles.

If it is assumed that $T_1 - T_2$ is zero for contact angle $\alpha = 0$, it would seem that the best compromise formula to suggest is of the form

$$T_1 - T_2 = k\sqrt{\frac{\alpha}{100}}(T_1 + T_2) \quad .. \quad (5)$$

where k is an experimental constant.

From the practical point of view a formula giving the effective pull $T_1 - T_2$ directly from the total pull $T_1 + T_2$ is decidedly preferable to a formula giving T_1/T_2. The use of T_1/T_2 as a belt drive characteristic springs merely from the theoretical formula, eq. (1), when wv^2/g is neglected.

Values of the coefficient of friction f in eq. (1) must be derived from tests which give $T_1 - T_2$ and $T_1 + T_2$ directly, and must be derived under assumption of a contact angle which may have little to do with the actual contact angle. However, as the author has in the past been an advocate of experimentally determined T_1/T_2 values as a belt characteristic, and this usage has found widespread acceptance, it may be well to compare T_1/T_2 values derived from eq. 5 with values now commonly used in industry.

It can easily be seen that

$$\frac{T_1}{T_2} = \frac{(T_1 + T_2) + (T_1 - T_2)}{(T_1 + T_2) - (T_1 - T_2)} = \frac{1 + k\sqrt{\frac{\alpha}{100}}}{1 - k\sqrt{\frac{\alpha}{100}}} \quad .. \quad (6)$$

Values of k_1 (for floating drive) and k_2 (for locked drive) for use in computation practice must, for the present, be settled by recourse to general practical experience. A designer who wishes to play perfectly safe even for small pulleys may take $k_1 = 0{\cdot}5$. With reasonably large pulleys k_1 values of 0·6 might readily be attained.

It was found that in only one case was k_2/k_1 less than 1·1, while in many cases it was considerably higher. Hence it is conservative to say that at least a 10 per cent higher effective pull can be transmitted with fixed shaft distance than with a floating drive, the initially set $T_1 + T_2$ value being the same; provided, of course, permanent stretch is still inconsiderable.

FLAT-BELT PULLEY CROWN: Pulley crown tests had been preceded by so-called crown-conformation tests. These indicated that wide modern belts, particularly with steel-cable transmitting elements, might not conform to the pulley rim at the edges, unless the crown was quite low. This, of course, would have a tendency to concentrate the power transmission of the belt toward its middle section only, which could not have a favourable effect on the life of the belt. The question then arose how much crown was absolutely necessary to prevent the belt from running off the pulleys.

Initial orientation tests showed that a crown of 0·06 inch on the diameter per foot of pulley width was sufficient to steer the belt, but only if the shafts were specially aligned for each belt. A crown of 0·10-inch per foot was necessary to guide all belts without special alignment.

A Mechanical Speed Converter

(From *Polytechnisch Tijdschrift, Holland*, Vol. 3, No. 47/48, November 30, 1948, pp. 754a-755a, 2 illustrations.)

A NEW mechanical transmission unit with infinitely variable speed has been developed by N. V. Hubro Elektrotechn. Bedrijf, of Breda, Holland.

The converter unit gives instantaneous and continuous speed control over 63 per cent of the entire speed range. The machine is a simple and compact design, approximately 23½ in. long and 20 in. high. Adjustment to any desired speed is obtained by turning a handwheel.

The speed reduction is obtained by means of a system of Vee-belts and pulleys (see Fig. 1). The pulleys are of the split type, each half-pulley being mounted on a separate shaft. The front half-portions of the pulleys are mounted on shafts which can be shifted axially, by operating the handwheel, so as to vary the gap between the two half-portions of each pulley. The Vee-belt then slips down into the wider gap, or is forced upwards to a position corresponding to a greater pulley radius, and the pulley ratio is thus varied according to the required speed.

The motor always runs at full speed. The internal reduction ratio of the two pulley systems can be varied from 1/1 to 1/2·7. Furthermore, the reduction ratio between the primary pulley system and the motor is also variable, and the speed ratio between the secondary pulley system and the driven machine can be adjusted from 1·1 to 1·5, so that a considerable range of speeds is available. The shafts are mounted in six radial and thrust ball-bearings.

In the present arrangement, the motor with a specially designed base is built integral with the converter unit, but a new model is now being developed which will be capable of operating with any type of motor.

The unit is solidly built and noiseless in operation. The belts can be tightened up in a few minutes, without any dismantling, this being necessary after a certain length of time owing to the "give" of the Vee-belt material with time. The transmission is claimed to be thoroughly reliable, and the belts were found to be in good condition even after the machine was used eight hours daily for over a year.

The speed reduction is indicated on a dial by a pointer, as a percentage of the nominal speed of the motor shaft. The unit can be set up either vertically or horizontally, or in a 45-degrees position on either side. The Netherlands patent for this speed converter unit has already been published, and patents have been applied for in other countries.

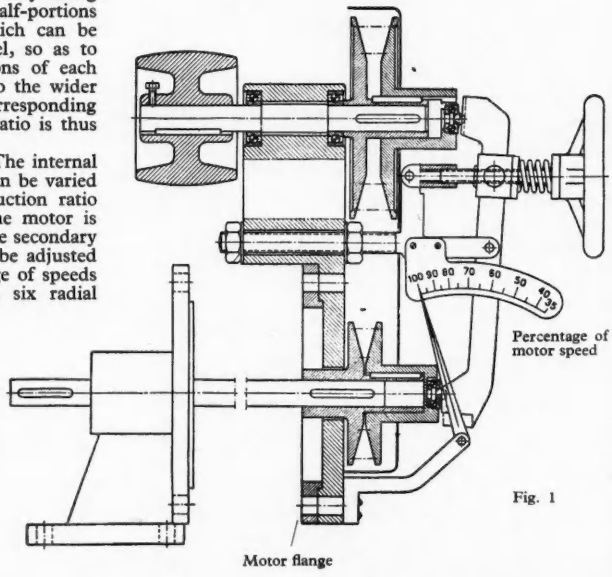


Fig. 1

PRODUCT DESIGN AND DEVELOPMENT

Some Notes on Product Design

INDUSTRIAL nations are now generally awakening to the growing importance of the design aspect in their plans to establish markets. Thanks to the efforts of the Council of Industrial Design, the British manufacturer is now fully aware of the need for good design in products for export. He also recognises that without a sound home market and an educated public appreciative of the fundamental attributes of good design, he cannot ensure development in the design of his products. On this understanding, the expression " Design for Exports " can be primarily construed as " Design for National Standards." The home markets of great industrial nations offer a vast testing ground for the selection of suitable products, and it is only when they have been approved in the home markets that they should be regarded as suitable products for export.

It is, therefore, imperative that we should aim at a technique and appreciation of design which combines fitness for purpose of the product and its aesthetic appeal to the buyer. In other words, we have to seek to embody in the product those qualities and features which arouse in the discerning buyer an acquisitive impulse towards the product. This, to a certain extent, involves the education of the buyer, because lack of good judgment may lead him to disregard bad design features which may render the product unacceptable on markets demanding high standards of industrial design.

Adherents of the old school still claim that engineering design has only one criterion, namely, fitness for purpose. The acquisitive impulse, they say, does not enter the argument as to whether and when a new machine should be bought. Obviously, this no longer holds good under present-day industrial conditions. If a new machine can be produced which takes up a smaller floor area than the old design, if the disposition of its working parts is conducive to better comfort for the operator, thus making for increased production, and if the colour scheme selected is one where the non-working parts are of a receding colour whilst the critical parts and handles are so coloured as to focus the workers' attention, with less eye-strain, on the operations to be carried out, the acquisition of the new machine can be economically justified long before the working life of the old machine is spent. Furthermore, a machine is not an isolated object ; it has to fit in with its environment. Modern shop lay-outs and lighting, in order to provide what is now termed " good environment for vision," call for the use of machines incorporating the features of progressive industrial design.

To some of the readers the term " industrial design " or " product design " may not convey a definite meaning, and, indeed, simple definitions have yet to be found in order to avoid ambiguity. To familiarise our reader with this subject, we feel we can do no better than refer him to a series of seven papers on industrial design recently read before the Royal Society of Arts.

These lectures stressed the unity of art and industry and, as we see it, brought out the need for a new type of designer. Hitherto, the designer was either of the " steam valve " variety (we prefer to call him " engineering designer ") or of the " life " variety. The latter we would call a " decorative designer " or " artist." We do not propose to concern ourselves with any disunity between the two and, in particular, we do not propose to go into the question as to which of the two should be called in by the manufacturer attempting to re-design his obsolescent product. We intend to judge the final product on its own merits irrespective of the group to which the designer appertains. Industrial designers are sometimes thought of as artists with an amateur knowledge of engineering and sometimes as engineers with artistic inclinations. However, we do think that for engineered products, and it is these which are to be reviewed in our new section " Product Design," the designer may be found to be primarily an engineer by education and experience, but one who, in addition, is critical on aesthetic matters and capable of learning as much as can be taught in art. He may conform closely to what we have called " the new type of designer." That the latter is clearly emerging is evident from the fact that two aircraft designers have been elected to the Faculty of Royal Designers in Industry. We regard this fact as pertinent because external form in aircraft is purely functional. In this instance, " fitness for purpose " cannot be achieved by aesthetic considerations However, generally speaking, the aerodynamically sound shape happens to give the greatest aesthetic appeal, just as though beauty in Nature were primarily due to functional perfection.

As to the external form of engineering products, the aim is to develop smooth continuous lines with a uniformly changing rate of curvature. The requirements of aerodynamics dictate minimum drag values for aircraft and many related fields of engineering endeavour. Thus, the development of shapes of aircraft, automobiles, projectiles and missiles, and shiplines, as well as product shapes in many other, similar or related, branches of engineering, demands more than a superficial interest in geometry. To avoid drag penalties, which are expressed in operating costs and performance, it is no longer sufficient to draw the curves " by eye." To keep the drag of any given shape as low as possible, these curves should be developed from analytical expressions so as to ensure uniform rates of change in curvature. These methods of analytically developing curves and streamlined forms have been employed for many years in the design of aircraft.* The requirements of shape in general product design are not as exacting as those in aircraft design and in all those fields of engineering where motion in a fluid or gaseous medium takes place. However, even in static products the analytical treatment of shape has many advantages. It permits the easy calculation of volumes, surface areas, etc., of such bodies, and enables quick changes to be made in shape, volume or area. This is of great value in developing subsequent designs from the prototype. Moreover, a firm reference basis is established for checking and measuring purposes, so that a high degree of uniformity of products can be maintained. This method has rendered all other methods obsolete, especially in fairing the intersection between two bodies. It consists in performing the work of generation purely analytically by representing various curves evolved by algebraic equations. The more general use of analytical methods would amply repay those who are earnestly concerned with Product Design. However, a detailed explanation of this method would be outside the scope of this article. **T.H.**

**The Analytical Development of Curves and Streamline Shapes*, by Harry H. Haase, published by Deemar Co. Amityville, L.I., N.Y., U.S.A., 1948.

Functional Design

"AN INTERESTING AMERICAN OPINION ON SOME PROBLEMS RELEVANT TO PRODUCT DESIGN."

By PROFESSOR EARL D. HAY, Iowa State College.

WITHIN the last twenty-five years, much has been written to make the public aware of the term "functional" as applied to design, whether it be applied to the design of buildings, furniture, household equipment or machines. It has been mentioned so often that it has almost become a catchword and a fad and sometimes used as an excuse for poor design, implying that a design is complete if it is functional.

Functional design is the oldest form of design and is rational in its concept. The first designs made by man were for definite utility, without embellishment, severe and crude, and useful for overcoming his enemies and in securing the means for his existence. It was only after life became something more than a battle for self-preservation and existence and some leisure time was found that primitive man began to ornament his tools and weapons with surface designs and colour, being careful that the embellishments did not weaken his weapons and thus detract from their functional use and perhaps endanger his life.

The distinction between functional and non-functional design as applied to machines may be made by a simple illustration: Suppose a non-functional designer is given a piece of wire; he may begin to bend the piece of wire in various directions in sweeping curves or sharp turns as fancy directs him, changing and modifying until the final result becomes a design unity, all parts in rhythmic and harmonious relation and the result satisfying to the eye. Then, after observing his abstraction from various angles, the designer may suddenly perceive that the result of his labours might have some practical use and exclaim, "By Jove, it could be used as a hatrack!"

Again, the functional designer, given a piece of wire, might set up a specification for a hatrack at the start. He would know before he made a single bend in the wire that the resulting design would be a more or less perfect hatrack. Before making the final design he would determine the functional requirements of a hatrack, its shape, the best proportions for strength and rigidity, simplicity of manufacture, etc., and, if he is a skilful designer, the final design will probably be pleasing to the eye.

In the first case the chances are very remote that the result will be a satisfactory hatrack. In the second case the result will be, without doubt, a hatrack meeting the specifications more or less perfectly, depending upon the skill and judgment of the designer.

The designers of machines, furniture, and household equipment have perforce followed more or less functional requirements through the ages, while the architectural designers, with less restrictions, went very far afield in blindly copying the temple designs of the Greeks and Romans and applying them to all sorts of buildings for which they were not functionally adapted. Greenough, one hundred years ago, began a crusade for functional design as a principle to be applied in the design of buildings, equipment, and machines. It was he who coined the phrase "Form Follows Function." His crusade was championed a generation later by the architects who had digressed much farther than the designers of machines from the functional approach to design. Louis Sullivan, the great exponent of "Form Follows Function" was followed by such architects as Frank Lloyd Wright and Eliel Saarinen in emphasising the functional approach to design, and, in recent years several of the industrial designers have adopted the form-follows-function phrase as their slogan for functional design.

The functional designers who insist that the phrase "Form Follows Function" provides the basis for the complete solution of the problem and that a machine so designed that it is functionally complete will, of necessity, have a pleasing appearance, would eliminate any need for the stylist who designs for appearance. However, there are many who will make a strong case for the stylist. The stylist differs from the above-mentioned non-functional designer, or abstractionalist, whose designs follow the laws of pleasing form and colour and whose designs *may* be found to have an utilitarian value, in that the stylist starts with a design which is more or less functionally perfect and, by applying the laws of art form to it, will not detract from its functional use, but make it more pleasing to the eye and more acceptable to the buyer or user. The functional design at least provides the anatomy or body upon which the stylist must do his hairdressing, manicuring, and draping to provide the functional machine with eye appeal of the highest order.

The problem of functional machine design as stated in the phrase "Form Follows Function" sounds exceedingly simple. State the function and the parts will assume appropriate form. However, the solution is usually far from being simple. Even the statement of the function of a complex machine is filled with many conflicting variables, the relative importance of these being given widely different values by different designers. If this were not so, all designs of such things as automobiles, stoves, refrigerators, and washing machines would be identical.

Because of the difficulty of properly evaluating the functions of a complex machine, many companies engage in consumer research, sending out questionnaires to secure information from prospective customers to be used as guides in determining the desired functions of their products. General Motors has been the leader in this kind of research in the automobile field.

In connection with consumer research in the automobile field it might be said that, if the functional design of the current models of many cars was based upon such research, the research problem set-up must have been very faulty indeed. Many who have used the current models will agree that in functional use the new models leave much to be desired and in many respects fall far below the preceding models in their functional requirements. In stating the case of the new model automobiles, George Cawley, the Design Consultant, states in the December issue of *Product Engineering*, that the cause of securing incorrect data from consumer research questionnaires lies in the statement of the questions. They are too much like the question put to a witness by a lawyer on the opposite side of the case. "Have you quit beating your wife? Answer yes or no." The answer in either case placed the witness in an unfavourable position and most likely did not bring forth a truthful reply.

In securing functional design data by consumer research questionnaires, the questions asked should not produce answers which are absolute, but answers of comparison or of relative values. For example, in securing data on consumer desires in automobiles, the question on the function of the seats should not be stated: "Do you desire wider seats in your next car? Answer, yes or no," but rather: "If you desire wider car seats, would you choose a car with wider seats if, in securing the wider seats, it becomes necessary to eliminate the wings and the usual body repairs become much more expensive, and if, should you park your car in a diagonal car park and cars of a similar style should happen to park on either side of you, you would not be able to enter your car until one of them moved away?" By securing questions with

relative answers, the importance of the various functional requirements may be ascertained. What the designer needs to know is the functions that are " musts" and those which are desirable, but relatively unimportant. It is very important that the designer should determine what are the essential functions of a machine, because, if the machine is not functionally acceptable, all of the styling which may be applied to it will not keep it on the market for any length of time. It will have the sales appeal of a well-dressed cripple at a fashion show.

Just what are the basic principles upon which a successful functional design is based ? First, and probably the most important, will be the correct statement of the desired functions of the machine followed by a statement of the relative importance or desirability of these functions. Without this statement, the designer will not be able to attack the problem intelligently. In the design of a complex machine, it would be a rare case in which the designer would be able to meet all of the desired functional requirements, because the requirements of some will conflict with the requirements of others. The less desirable will have to give way to the more desirable. The design of an electric refrigerator of a given storage capacity supplies a simple example. The more the space allotted to ice cube freezing, the less the space available for food storage. It is the designer's task to place these functions in proper balance, and, to make the task more difficult, the ratio does not remain the same for all sizes of refrigerators. A study of consumer needs as well as consumer desires as determined by research will provide a guide or trend, but, in the final analysis, much will depend upon the judgment of the designer. Moreover, as designers vary in judgment, so do we find that machines built by different companies to perform supposedly identical functions vary considerably in functional performance and use.

After a clear and complete statement of the functional requirements of the machine has been secured, the designer is ready for the second step. He must begin to make selections. He must first select a mechanism or mechanisms which will produce the functions the machine is to perform. He will probably be able to find several mechanisms ranging all of the way from mechanisms of great complexity to a mechanism of marked simplicity, that is, if he searches long enough. Some will require a great deal of space, some little ; some will be easy to manufacture, others costly. The designer must now select the materials for the various elements of the mechanism. This selection has become a very complicated and involved problem, as the number of materials for machine manufacture has grown from a dozen or two to several hundred in the last fifty years, and the number of processes by which they may be prepared also runs into the hundreds. Next in order, the designer will apply the principles of machine design and begin to give the various elements of the machine form and substance in order that they may adequately and safely carry and transmit the various loads. In this process he will probably need to make many changes and adjustments from his first selections because of space requirements and interferences in order to make the machine a workable unit.

It is in this part of the design process that the stylist should be working with the functional designer. Form follows function, and, form somewhat modified or presented in a different material or finish may still follow function and present added eye or sales appeal.

As time goes on, the province of the functional designer becomes more complicated and extended. It is now practically impossible for one man to keep abreast of the developments in this field. The field is being divided into smaller areas in which specialists are being developed. Specialists in consumer research, whose province it is to find the complete and accurate statement of the problem. Specialists in mechanism selection, highly trained in kinematics, dynamics, and vibration analysis. Specialists in the properties of manufacturing materials, physical metallurgy, surface finishes and manufacturing processes. Specialists in basic design, the accurate determination of loads, stresses and strains in statically determinate and indeterminate structures and in the determination of the most economical form for the various machine elements, and, most recently, specialists in control mechanisms and methods, instrumentation, and electronics applied to machines.

The field of functional design continues to expand somewhat, but its greatest recent growth has been in its internal complexity. In this direction, the end is beyond the limits of the most vivid imagination.

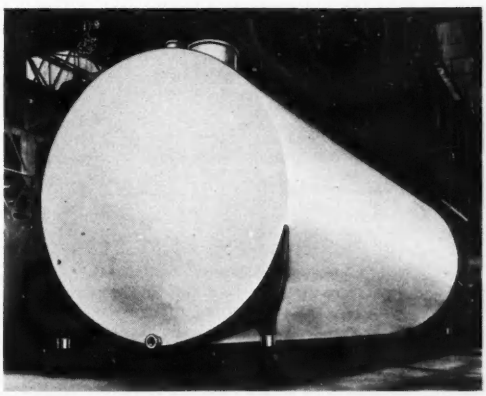
Some of the machines and devices shown in the following pages have been selected to demonstrate the harmony between appearance and functional objectives.

They serve as illustrations of some of the factors discussed in the introductory articles of this new Section and give an indication of the wide range of products with which the industrial designer is concerned.

* * *

GLASS-LINED CHEMICAL STORAGE TANK

The Pfaudler Co., Rochester, N.Y., announces the production of one-piece glass-lined chemical storage tanks in greatly increased sizes up to 35,000 U.S. gallons capacity. Long needed by the Chemical Industry for bulk storage of chemicals, the successful engineering and ceramic developments which went into producing these large glass-lined chemical storage tanks will eliminate frequent and costly replacements of other materials of construction due to corrosive conditions present.



Design features of the tank supporting legs are noteworthy and are a good solution where a bulky and heavy container has to be carefully handled owing to the fragility of its lining.

PHILIPS ELECTRON MICROSCOPE

The Philips Electron Microscope (Type 11980) incorporates many novel features previously not available in Electron Microscopes of conventional design.

The instrument, which is of the magnetic type, has, in addition to the normal set of three lenses, that is, condenser, objective and projector, two extra lenses fitted, which makes possible a variable magnification covering the whole range from 1000-60,000 diameters without the necessity of changing the pole pieces. Further, by the use of a wide angle projector lens operating at a constant strength, coverage of the full screen area of 8 inches at all magnifications is attained.

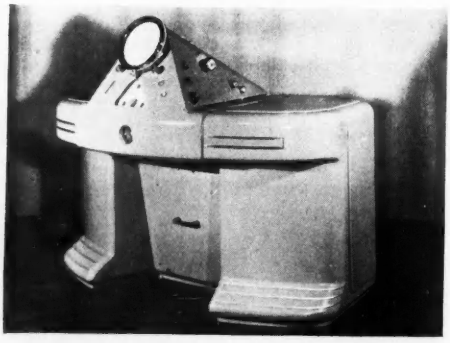
The design of the microscope is such as to render adjustment of the alignment of the lens system unnecessary. Adjustment of the illuminating field is effected simply by means of electrically controlled magnetic deflecting fields, thereby obviating the necessity of effecting mechanical adjustment.

Full provision is made for obtaining Stereo-Micrographs, by rotation of the specimen holder between two successive exposures. Simplicity of specimen mounting is assured by the introduction of a special specimen carrier.

The photographic arrangement is such as to permit the use of an internal camera employing standard 35 mm film capable of recording the screen image at a reduced scale of 4 times. Up to 40 exposures can be made without reloading. An automatic film shift after each exposure prevents the possibility of double pictures.

The Philips Microscope is not sensitive to vibrations and no special foundations are required for its mounting. All apertures are easily removed for cleaning without dismantling the microscope assembly, and the whole equipment is completely shockproof and suitable for operation in any climate as well as at any altitude.

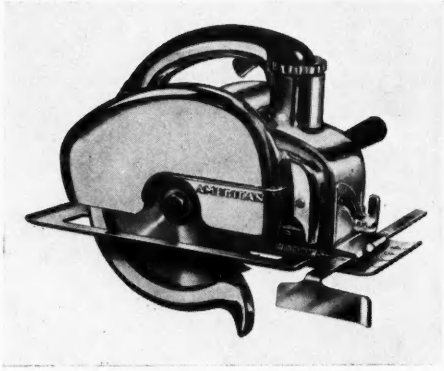
From the viewpoint of functional and aesthetic design, this electron microscope should satisfy the most rigorous industrial design requirements.



PORTABLE ELECTRIC SAW

A good example of industrial design applied to a tool for all-round use is the new American saw manufactured by the American Floor Surfacing Machine Co., Toledo, Ohio, which has an 8½ in. blade. The motor, which develops 2⅛ hp, supplies the power and capacity needed for the handling of everyday jobs in cutting wood, sheet metal, stone, tile or compositions. The saw will cut a 2⅛ in. plank at 45 degrees, or will make a straight cut of 3 in. The tool is balanced for one-hand operation, and a balanced torque principle, in which the motor and saw blade turn in opposite directions, keeps the saw from jolting or twisting when starting.

Extra strength and stability are provided by a wide base of hard aluminium alloy, 6¾ in. × 15⅜ in., with bevelled edges for smooth operation. Both sides of the base are accurately machined so that they are parallel with the blade, and either can be used against a guide strip in ripping. In regulating depth adjustment, the handle is always in the same relative position to the base.



The ripping guide is accurately calibrated. In addition, the front of the base plate is calibrated in inches, so that many kinds of guide lines may be used, either to the right or left of blade. American double post construction, allowing positive adjustment with triple ACME lead thread, makes it possible to adjust the blade for any desired cut without variations, once it has been set.

Other features include a telescoping blade guard, life-time lubricated bearings, and totally enclosed trigger-type safety switch on handle. The saw weighs 19½ lb. The frame is of high-quality cast magnesium alloy, buffed to a satin finish. Standard equipment includes a metal carrying case, wrench for removing saw blades, and a combination rip and cross-cut blade.

HIGH VACUUM PLANT

A laboratory vacuum pump plant, Balzers LAB 300/1, manufactured by Etablissements de Constructions d'Appareils Balzers, Liechtenstein, meriting attention by its neat arrangement and smallness of dimensions accomplished, is shown in Fig. 1. It deviates from customary practice inasmuch as the time of experiments is not determined by the equipment. The pumping speed is such that a vacuum system of 50 litres content can be brought from atmospheric pressure to a vacuum of 5×10^{-5} mm mercury within five minutes. It is the new design of the valves that

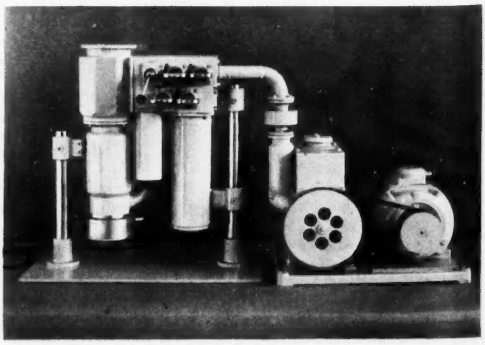


Fig. 1

gives this plant a character of its own. All valve members, vacuum distributions and vacuum gauges are enclosed in a compact valve block. Four of these valves are operated by drawer pull-type handles, while the high-vacuum valve is operated by a lever with a ball handle (Fig. 1).

The function of the plant is explained by reference to the schematic illustration (Fig. 2). The rotary roughing pump VP with a pumping speed of 15 m^3/hr and a limiting pressure of less than 10^{-3} mm mercury is connected with the valve block through damping member DG and pipe VL. Pump VP can be connected directly through valve II with the vessel E to be evacuated (E is omitted from Fig. 1), when high-vacuum valve I remains closed. This is the arrangement for roughing; for high vacuum, pump VP is connected through valve III with the low vacuum container VB

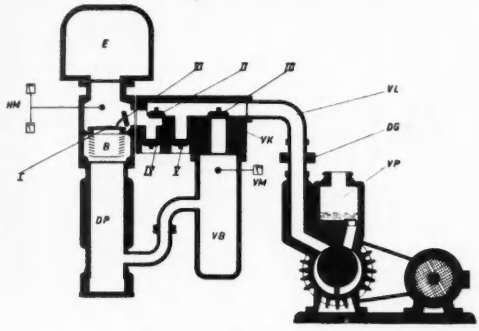


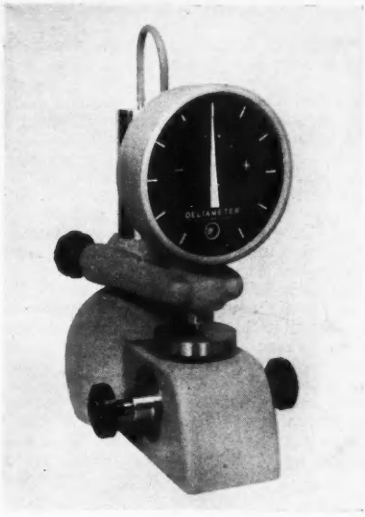
Fig. 2

and the oil diffusion pump DP, valve I being open. This automatically shuts off all other valves from the high-vacuum side through valve VI. Valve IV serves to flood vessel E (when valve I is shut), the purpose of valve V being to flood the roughing pump VP and thus put it out of operation. The water-cooled baffle B is directly above the oil diffusion pump. The limiting speed of the latter is 300 litres per second at 5×10^{-4} mm mercury. The attainable limiting pressure is less than 10^{-5} mm mercury. The vacuum is measured at VM and HM.

PNEUMATIC MICROMETER FOR AUTOMATIC CONTROL

Pneumatic micrometers of varying design have already been described in scientific publications. A micrometer, called the Deltameter, working on a principle quite different from these, has been designed by C. G. Hard of Segerstad, chief engineer in charge of engineering research, The Sandvik Steel Works Co., Sandviken, Sweden. The instrument has been used since 1943 partly as a comparator and partly as a primary member in connection with automatic control, automatic sorting, warning signals, etc.

The standard instrument has a scale of 300 degrees divided into 10 sections, each of which represents 1/1000 mm. Each division on the dial at the pointer tip represents 32 mm, thus giving a magnification of 32000, the total measuring range being 10/1000 mm. On the meter there is a lever by means of which the magnification can be decreased by half, making a magnification of 16000, and a total measuring range of 20/1000 mm. All instruments are equipped with two individually adjustable silver contacts, one for the minus side and the other for the plus side, connected to a 3-pole plug for make-and-break control. Depending on satisfactory torque, the instruments can also be equipped with a micro-potentiometer which, driven by



the pointer axis, can be used for proportional control, telemetering, etc.

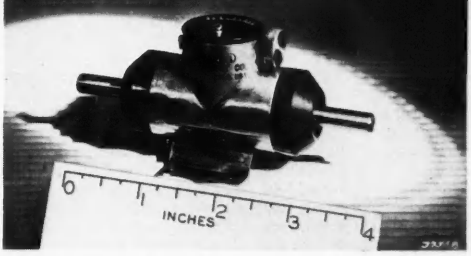
Ease of reading determined the dial size, which is also in harmony with the other dimensions of the instrument. A well designed product.

MINIATURE ADJUSTABLE-SPEED DRIVE

New design possibilities are created by the introduction of the Metron Miniature Variable-Ratio Speed Changer. Hitherto, adjustable-speed low-power drives were often ruled out due to their large size and heavy weight. This small compact unit weighs less than six ounces, and delivers up to 70 ounce-inches torque and 0·025 horsepower. Possible applications include business machines, recorders, controllers, calculators, indicating mechanisms and similar devices.

The ratio of input to output speeds is infinitely adjustable between 1 : 6 increase and 6 : 1 decrease, with a total speed range of 36 : 1. A dial and pointer indicate ratio setting and the adjusting knob is equipped with friction drag to prevent radio wander. Maximum output torque is obtainable at zero speed, and operating speeds as high as 20,000 rpm are practical.

The internal construction employs rollers, which make contact with discs on the input and output shafts.



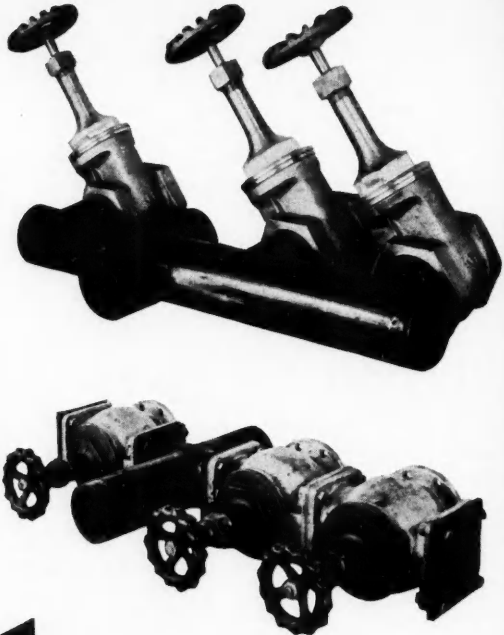
Speed is adjusted by changing the radius at which the rollers make contact. Novel "lock-up" devices are provided to increase contact pressure automatically when the load torque is increased. This ensures high maximum output torque and minimum reduction in speed with increased load, and at the same time is subject to low loss when lightly loaded.

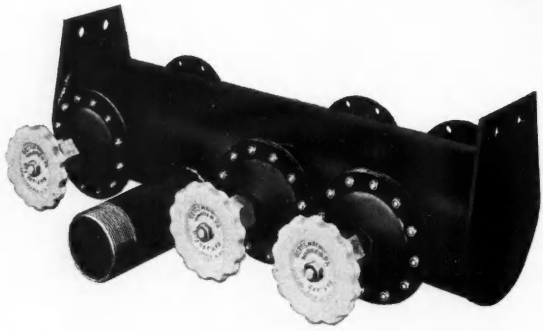
MANIFOLD VALVE

After years of development work on manifold valves, the Betts Machine Company announce their latest product. This special manifold valve permits free-flowing of oil and other liquid materials, with less loss of head, and appears to be a highly efficient valve.

The new "Warren Manifold Valve," on which patent applications have been filed, is constructed from seamless steel tubing and stampings, allowing valve arrangements to be confined to the smallest possible space.

The photographs illustrate the evolution of the manifold valve and, for purposes of comparison, shows the costlier and less efficient earlier types. Industrial engineers who utilize manifold valves in the manufacture of their equipment can easily recognize the feature of larger flow areas, permitting faster flow of materials. There is no risk of leakage, because of the judicious use of steel tubing and stampings. It will also be noted that the new valve not only has cleaner lines, but is also "streamlined," and, therefore, takes up less room than old-styled units. Furthermore, valve arrangements are limited only by the designer's ingenuity, as a manifold valve is now available which can be easily cut and welded to meet any requirements. No doubt, this product will stimulate creative thought in engineers and designers.





OLD DESIGNS (above)
NEW DESIGN (left)

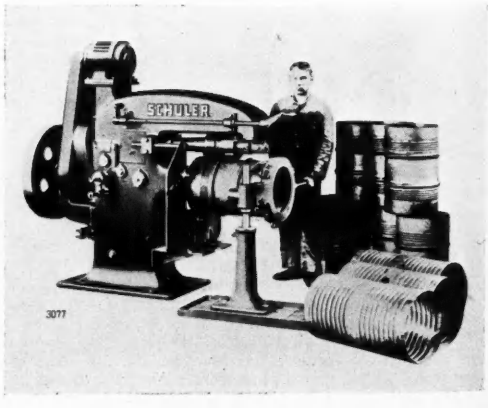
DRUM-MAKING MACHINERY

Economical metal drum-making machinery, capable of fabricating metal of up to $\frac{1}{16}$ in. thickness, is manufactured by L. Schuler, A.G., of Goeppingen, Germany.

The Schuler machines have been specially designed for quick change-overs from one drum size to another. The equipment is also adaptable for installation in a conveyor line for producing large quantities of a single size.

The beading machine shown in the illustration, used to press the rolling and stiffening beads, is semi-automatic. The drum body is held in position between two centering rings while one half of the drum is worked. One would least expect to find beauty in this kind of machinery, but this beading machine is well-built, and looks good. It can handle drum bodies with diameters from 12 to 24 inches and heights up to 43 inches. The Schuler firm also manufactures a second type of beading machine, in which the beads are all rolled in one operation. This machine handles barrel bodies from 10 to 28 inches in diameter and up to 39 inches in height. The action of the beading rolls is such that only one bead is formed at a time, so that the metal is pressed uniformly without the formation of any thin spots.

A double flanging machine produces a flange on both ends in one operation, and a third machine, for seaming operations, accomplishes the setting of the top and bottom lids.



CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

CARBON MATERIALS

Carbon-Graphite Parts

By FREMONT F. RUHL. (From *Product Engineering, U.S.A.*, Vol. 19, No. 2, February, 1949, pp. 81-85, 11 illustrations.)

POWDERED carbon and graphite, mixed with a suitable petroleum base binder, compacted under high pressure and then furnaced at a temperature exceeding 3,000° F, produces a hard, strong and light material that possesses many unusual properties highly desired for certain applications. The final product is 95 to 99·8 per cent carbon, the binder having been reduced to carbon in the furnacing process. However, during processing the properties of the amorphous graphite are not measurably changed, the surface of the finished part possessing the slipperiness characteristic of that material which gives carbon-graphite an exceptionally low coefficient of dry friction. In combination with this unique property, carbon-graphite parts exhibit the characteristic properties of carbon. The material has no measurable plasticity or ductility even at elevated temperatures, and its coefficient of thermal expansion is about one-fourth that of steel. Hence carbon-graphite bearings or other parts undergo no appreciable change in shape or dimensions when their temperature is increased from room temperature to 600° F or more. In addition, the material has a relatively high heat conductivity, high heat radiating capacity, relatively high strength, unusual resiliency, and high hardness. This unusual combination of properties makes carbon-graphite the first choice material for many bearing applications and parts which are subjected to rubbing contact with metal.

Because of the many variables which enter into the manufacture of carbon-graphite to its finished form, the numerical values of the properties of the material are different for different grades and makes. The purity of the base materials, the fineness of the powders, the efficiency of the mixing, the compacting pressure, the furnacing temperature and many other factors appreciably affect the properties of the final product. By proper techniques and controls, innumerable grades of carbon-graphite have been produced, each grade tailored to possess certain unusual combinations of properties.

A general misconception is that carbon-graphite parts are highly brittle. On the contrary, balls when dropped upon a smooth concrete slab will rebound up to 73 per cent of the height from which they were dropped.

The article then deals with design precautions and recommends tolerances based on practical experience. The most important application of carbon-graphite is that of bearings, a series of graphs being given which may assist the designer and manufacturer in developing such bearings. They show, for instance, clearances for journals of various diameters at given temperature rises, permissible unit bearing pressures on bearings of various journal diameters and speeds, the wall thickness of the bearings for various outside diameters and lengths, etc. Other parts for which carbon-graphite is used are :—rotary pressure joint seals for handling steam, water, powdered talc and many chemicals, turbine packing rings, cylinder liners, etc. Other applications include vanes for rotary pumps, centrifugal pump wearing rings, moulds for the glass industry, pump valves, and metering discs. Summarising, it can be said that carbon-graphite can be successfully used in cases where rubbing surfaces cannot be lubricated for some reason or other, where the part will be subjected to high temperatures or to corrosive chemicals or media, and where high dimensional stability or high wear resistance is necessary.

ELECTRIC TRANSMISSION AND DISTRIBUTION

Constant Power Transmission over Long Distances

By N. SEIDNER. (From *Elecktrotechnika, Hungary*, Vol. 41, No. 3, March, 1949, pp. 61-66, 2 illustrations.)

THE author of " Energiewirtschaft " considers the basic principles of the co-ordination of energy production to achieve the lowest cost. Hydraulic plants work most economically with load factors which are well below unity. Transmission lines, on the other hand, are cheapest to operate when the load factor is unity. The cost of energy produced by steam plants depends, again, on the cost of transport of coal. These different factors must be taken into account when planning the provision of power for an industrial centre. The most economical combination for this purpose uses hydraulic power for the basic load, which means that the transmission line works at constant load. The peak load is provided by a steam power plant situated near the consumer. This steam plant works in co-operation with a hydraulic or electrical storage plant. The surplus power of the water plant should be used locally. The author calculates the economics of the energy transmission from the St. Lawrence River to New York. He finds that the cost of power is the lowest in New York, if the power is provided by the combination of producers just described. The cost is 23 per cent lower than in the case of all the energy coming from a steam power plant located in New York. The author applies these principles to power planning in Hungary.

SCIENTIFIC INSTRUMENTS

A New Direct-Reading and Recording Apparatus for Spectrographic Analysis

By P. COHEUR and A. HANS. (From *Revue Universelle des Mines, Belgium*, 9th Series, Vol. 5, No. 2, February, 1949, pp. 63-68, 9 illustrations.)

ELECTRONIC amplifiers and a camera can be incorporated in a photo-cell spectrograph and analyses can then be made, either simultaneously or alternately, from direct readings or recorded data. Results obtained from such an installation, built by the Applied Research Laboratories, show that analyses can be made very rapidly and accurately. The mean quadratic error of a direct analysis of steel could be shown to be less than 1 per cent (0·6 to 0·8 per cent), and the time needed for direct analysis could be shown to be 55 seconds for the first element and 5 seconds for each additional element. The time taken for a recorded analysis was 5 minutes for the first element and 20 seconds for each additional element, and the mean quadratic error was of the order of 1·5 to 2 per cent.

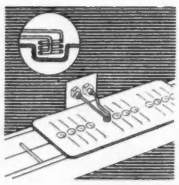
The spark rays pass through slits and are reflected by concave mirrors onto photo-cells. The photo-cell

a better way to do it

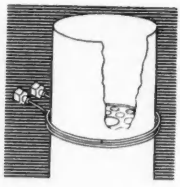
How about that awkward production job — that tricky bit of brazing or annealing, or that next to impossible soldering operation? The chances are that Redifon Heating will do it more quickly, cheaply, and much better than any other method.

The Redifon IH.38 is an Industrial Radio Heater giving an output of 1.kW, and is specifically designed for long trouble-free service on hard, slogging mass production tasks.

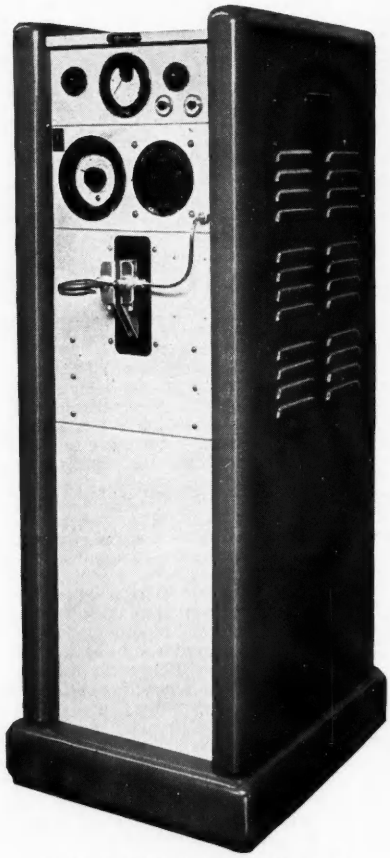
Our Industrial Application engineers will be pleased to give impartial advice on any production problem involving soft-soldering, brazing, hardening or annealing of small parts.



JOB: Local annealing stainless steel deep-drawn pressing.
GENERATOR: IH.38 at full power.
MATERIAL: Austenitic stainless steel approximately 16 SWG.
TIME: 10 seconds per position.



JOB: Soldering of baffles in vacuum cleaner body.
GENERATOR: IH.38 at full power.
MATERIAL: Tinplate, solder.
TIME: 20 seconds.



REDIFON IH.38
INDUCTION HEATER

Redifon HEATING

INDUSTRIAL ELECTRONICS DIVISION
REDIFFUSION LIMITED, BROOMHILL ROAD, WANDSWORTH, S.W.18
Designers and Manufacturers of Industrial Electronic and Radio Communication Equipment

current charges capacitors, the voltage of which is measured in high-impedance pentode bridge circuits with a potentiometric registering device in the diagonal arm. Zero shift is controlled at the grid of one of the pentodes, and sensitivity by a shunt in the potentiometer branch. Two standard samples, one with a high concentration, another with a low concentration of the desired element, are used to charge two capacitors which can be alternately connected to the bridge circuit, so that two scales of concentrations can be used. The grid control of the second pentode automatically compensates for any shunting effect, so that the scale chosen need not be corrected. The sparking is interrupted as soon as the registering device reaches a certain voltage, and the length of sparking periods can thus be controlled by the shunt. A small lamp of automatically adjusted brilliance is switched on before sparking begins, so that the photo-cells are adjusted to the brilliance of the sparks. When sparking begins, the lamp is switched off and the capacitors switched into the circuit. The capacitances automatically integrate the light intensities over a period of, say, 30 seconds. If, however, the variation of light intensity is required to be registered, the capacitance can be replaced by an impedance. This permits adjustment of the spectrograph slits to an optimum position. The spectrograph itself is equipped with a grid of 24,400 lines per inch and has been adjusted, for practical analyses, to register wavelengths of 2475-3725 Angstrom for direct reading, and 4950-7450 for recording.

By comparison with the measured standard brilliance of the small lamp, the accuracy of the instrument was found to be within the reading error, i.e., 0·1 per cent, whilst the remaining 0·5 to 0·8 per cent is due to fluctuations in the light source. Results within this accuracy are reproduceable. The difference in mean quadratic error between graphic and direct-reading analysis is ½ to 1 per cent. By virtue of its accuracy and speed of direct analysis, this instrument is, therefore, excellently suited for production control. If used in research, it facilitates determination of optimum conditions for research analyses and elimination of possible sources of error. As a recorder, the instrument is suitable for qualitative or semi-quantitative analyses or for particular quantitative measurements.

SPRAY COATING

Steel Spraying on Aluminium

(From *Revue de L'Aluminium, France*, Vol. 25, No. 149, November, 1948, p. 361.)

A RESEARCH Committee set up in 1944 by several interested French Technical Institutions studied the problem of steel spray-coating of aluminium and magnesium alloys and arrived at the following practical conclusions: Widely differing steels, from mild steel to high-carbon steels, 18/8 stainless and 13 per cent chromium steels, can be sprayed on to light alloys. Adhesion is obtained by mechanical interaction in the case of products with convex or cylindrical surfaces, such as axles and pulleys. For the spraying of concave surfaces, the product should be pre-heated so that, after cooling, it exerts a shrinkage pressure on the deposited layer.

The deposited metal is harder than the original steel used for spraying, owing to rapid cooling from the melt and cold-working during impact. The porosity of the deposited layer gives it self-lubricating qualities and the coefficient of friction is lower than that of rolled steel of identical composition. In spite of the porosity, a layer of 0·04 inch thickness is sufficiently air-sealing to prevent corrosion by electrolytic action. Where a light alloy product is only partially covered, a band of sprayed zinc will protect the edges of the layer against electrolytic action. Mild steel deposits can be machined with tools made from high-speed steels, having the same cutting angles and feed as are used for cast iron. Deposits of 0·6 per cent carbon steels can be machined with tungsten carbide tools.

Steel spraying gives light alloy products a wear resistance comparable to that of solid steel. This new technique is particularly advantageous for light-weight construction and in cases where light alloys could not otherwise be used owing to their inadequate wear resistance.

THERMODYNAMICS

Thermodynamic Analysis of Centrifugal Compressors

By L. G. VALDENAZZI. (From *Bulletin Technique de la Suisse Romande*, Vol. 75, Nos. 5 and 6, February and March, 1949, pp. 53-59, 65-69, 11 illustrations.)

CALCULATIONS of centrifugal compressors are normally undertaken in two stages. In the first, it is sought to determine the conditions of the fluid in the various elements of the compressor. To this end, the known laws of thermodynamics are applied and constant values are assumed for the efficiencies, based either on experiments or on past experience. The final aim of study in this field is to be able to trace the probable course of the compression on a Mollier diagram or the like.

In the second stage, it is attempted to determine from aerodynamic considerations the shapes and dimensions of the elements of the compressor, the results of the first stage of calculation also being used. In general, these two stages of calculation are by no means independent, but the assumptions entering into the calculations are corrected by trial and error in successive series of approximations.

The technical literature regarding such calculations is rather incomplete, and this is especially so in the case of cooled centrifugal compressors. Some authors are content to trace the "probable" course of the compression on temperature-entropy diagrams or Mollier diagrams in a manner dictated by their personal experience. This requires large numbers of trial calculations and, in any case, confines the use of such methods to engineers with extensive experimental data at their disposal. Other authors, following a very different trend of thought, have developed very complicated mathematical methods based on the heat transfer in the passages of the compressor. These require simplifying assumptions concerning both the shape of the passages and the flow coefficients. This approach thus suffers both from the difficulty of its application and the uncertainty of its results. To give the order of magnitude of the probable errors, it is only necessary to remember that the best authorities differ by multiples of 10 per cent in the values they assign to the flow coefficient in the very simple case of a cylindrical tube heated by the passage of warm air.

In view of this position, the author of this article has set himself the task of finding a thermodynamic method of calculation which should not only be simple and conform to practical experience, but also be based on a reasonable number of coefficients, which should be known or easily determined by experiment. In the article, careful analysis leads to formulae which can be evaluated in terms of dimensionless numbers (in the nature of efficiencies) describing the deviation from the simple case of isothermal compression. It is shown that the ideal course of compression as well as the deviations from it can be determined by simple aids, and examples are worked to show that only few trial calculations are required for satisfactory accuracy.

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NEWS OF THE MONTH

PERSONAL

Mr. R. Salter Bache, chairman of Geo. Salter & Co., Ltd., West Bromwich, has been elected President of the West Bromwich and District Manufacturers' Association and also President of the Coil Spring Federation of Great Britain.

Mr. W. C. Bennett has been appointed purchasing manager of Standard Telephones and Cables Ltd., Oakleigh Road, New Southgate, London, N.11.

Mr. R. Braid has been elected chairman and managing director of Braid Bros., Ltd., manufacturers of aircraft components, Colwyn Bay, N. Wales.

Sir John S. Buchanan, C.B.E., has been elected President of the Royal Aeronautical Society. **Mr. W. G. A. Perring, Mr. N. E. Rowe** and **Dr. H. C. Watts** have been elected Vice-Presidents.

Mr. O. V. S. Bulleid, C.B.E., M.I.Mech.E., has been appointed Consulting Mechanical Engineer to Coras Iompair Eireann.

Mr. F. Caffey, general manager of Darwins Ltd., Toledo Works, Sheffield, has been elected to the board of the Company.

Mr. C. H. Claxton has been appointed director of Thomas Hill-Jones Ltd., Invicta Works, Bow Common Lane, London, E.3.

Mr. F. I. Connolly has been re-elected President of the Society of Motor Manufacturers and Traders Ltd.

Mr. J. A. Dale has been appointed director of Kay & Co. (Engineers) Ltd., Bolton.

Mr. James F. Douglas, manager of the repairs department of Cammel Laird & Co., Ltd., Birkenhead, has been appointed local director of the Company.

Mr. E. F. O. Gascoigne has been appointed director of Cowlishaw, Walker & Co., Ltd., engineers and ironfounders, Stoke-on-Trent.

Mr. R. Griffiths, M.Sc., F.I.M., has been elected President of the Swansea and District Metallurgical Society for the 1949-50 session. **Mr. H. A. Cooke** and **Mr. M. J. A. Thomas, B.Sc.,** will be joint honorary secretaries.

Mr. W. Hilton has been appointed purchasing officer to the London Transport Executive, 55 Broadway, London, S.W.1.

Group Captain R. C. Hockey has been appointed aviation divisional engineer of the Cheltenham factory of Smiths Aircraft Instruments Ltd.

Mr. A. H. Johnson has been elected chairman of British Steel Constructions (Birmingham) Ltd., Wednesbury, Staffs.

Mr. James Leek, managing director of the Birmingham Small Arms Co., Ltd., Small Heath, Birmingham, has been elected President of the Birmingham Chamber of Commerce.

Mr. W. W. MacArthur, general works manager of Craven Brothers (Manchester) Ltd., has been elected a director of the Company.

Mr. A. W. Manser has been appointed to assume control of the Acton works of the London Transport Executive with the title of works manager (railways).

Mr. H. Mills has been appointed supervisor of the branch offices and of the exhibition organisation of Brooks Motors, Ltd., Empress Works, Huddersfield. **Mr. A. Schofield** has been appointed manager of the Newcastle branch in succession to Mr. Mills.

Mr. L. P. Parker, M.I.C.E., M.I.Mech.E., has been appointed Motive Power Superintendent, Eastern Region, British Railways.

Mr. Basil Sanderson has been re-elected chairman of the Shipping Federation, 52 Leadenhall Street, London, E.C.3. **Mr. D. F. Anderson** and **Mr. B. Eliot Common** have been elected vice-chairmen.

Mr. L. Turner, M.I.C.E., has been elected President of the Institution of Structural Engineers.

Mr. George R. Ward has been appointed director of Round Oak Steel Works Ltd., Brierley Hill, Staffs.

Mr. A. Vernon Watson, A.I.Mech.E., A.M.I.P.E., F.R.S.A., has been elected President of the Institution of Engineering Inspection.

Mr. J. F. R. Wiggins, M.Inst.T., Mersey Docks and Harbour Board, Liverpool 3, has been appointed honorary secretary of the Merseyside and District Section of the Institute of Transport.

OBITUARY

Mr. R. J. M. Whibley, sales manager of The Churchill Machine Tool Co., Ltd., Broadheath, died suddenly at his home in Hale Barns, Cheshire, on May 30th, at the age of 64. He was an Associate Member of the Institution of Mechanical Engineers and Member of the Manchester Association of Engineers. In 1947 he was awarded the Constantine Gold Medal for his paper on "Production of Flat Surfaces."

Mr. Whibley was well known in engineering circles, particularly in the Machine Tool Industry. Until 1926 he was on the editorial staff of "Machinery" and, for the past five years, Member of the Editorial Advisory Board of "The Engineers' Digest."

Mr. Robert Ernest Gale, manager of the High Frequency and Instrument sections of Philips Electrical Ltd., at the Aboyne Road works, Tooting, died after an operation in Guy's Hospital at the age of 47.

BUSINESS NOTES

Air Control Installations Ltd., Ruislip, Middlesex, has opened a new branch office in Manchester. **Mr. F. Wright, A.M.I.E.E., A.M.I.H.V.E.,** has been appointed branch manager. Enquiries at present should be addressed to Mr. Wright at 1 Vale Drive, Prestwich, Manchester.

The ALAR (Association of Light Alloy Refiners) development offices have been moved to 3 Albemarle Street, London, W.1. Tel.: MAYfair 2901.

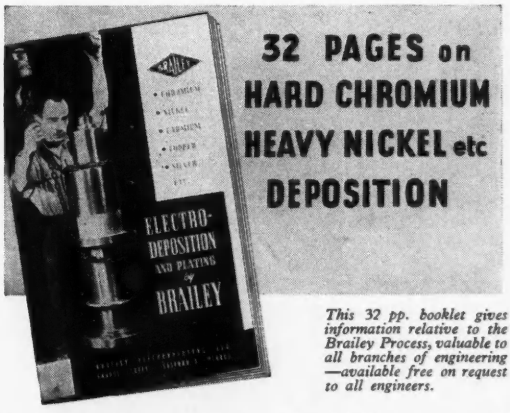
Ferodo Ltd. has moved its Belfast office to Ferodo House, 41 Linen Hall Street, Belfast. Tel. 24976. **Mr. F. T. Collett** is branch manager.

Freeman, Fox & Partners have been appointed consulting engineers for the construction of the Exhibition to be held on the South Bank of the Thames in connection with the Festival of Britain 1951.

The Metropolitan-Vickers Electrical Co., Ltd., has moved its London offices to St. Paul's Corner, 1-3 St. Paul's Churchyard, E.C.4. Tel.: CITY 5757.

J. Parkinson & Son (Shipley) Ltd. has been registered as a private limited company to carry on the business of J. Parkinson & Son, Shipley, Yorkshire. **Messrs. Ernest, Edward and Harry Parkinson** will serve as directors of the company.

Electronics Centre. Cinema-Television Ltd. and Dawe Instruments Ltd. have opened a new Electronics Centre at 83 Piccadilly, London, W.1, which will serve as permanent showroom and technical advice bureau on electronics. They hope to do useful work in the showrooms in exchanging information on the use of electronics in science and industry. Instruments on display include Atomic equipment; Measuring instruments; Stroboscopes; Industrial counters; Oscilloscopes; Communication equipment; Metal detectors; Photographic flash units; Photo-electric cells and Cathode ray tubes. The showrooms will be open daily, Monday to Friday inclusive, from 9 to 12.30 a.m. and 1.30 to 5.30 p.m.



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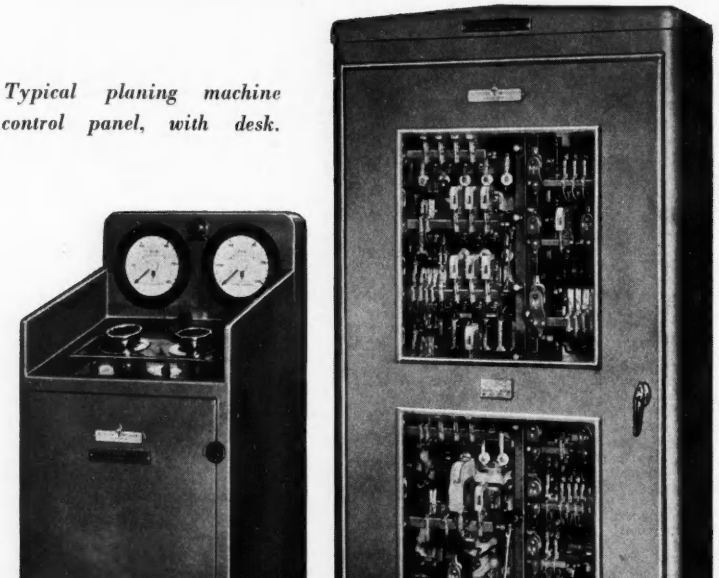

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MISSION OF THE MICHELSON LABORATORY

By L. T. E. THOMPSON

COMPLETION of the Michelson Laboratory at the Naval Ordnance Test Station, Inyokern, China Lake, California, has added an important link to the chain of research and development facilities which support the preparedness program of the United States. The development centre at Inyokern is operated by the Bureau of Ordnance of the Navy Department for the purpose of conducting research, development, test, and training activities necessary in meeting the requirements for new weapons. Rear Admiral A. G. Noble, Chief of the Bureau of Ordnance, stated at the dedication of the Michelson Laboratory on May 8, 1948, that " The Navy's basic problems in maintaining a position of material advantage as the dominant world sea power are currently believed to be :

1. Increased air striking potentials.
2. Increased anti-aircraft effectiveness.
3. Increased anti-submarine effectiveness.

These most important broad naval problems, therefore, dictate the emphasis on the Bureau's present three programs—the aircraft program, the anti-aircraft program, and the anti-submarine program."

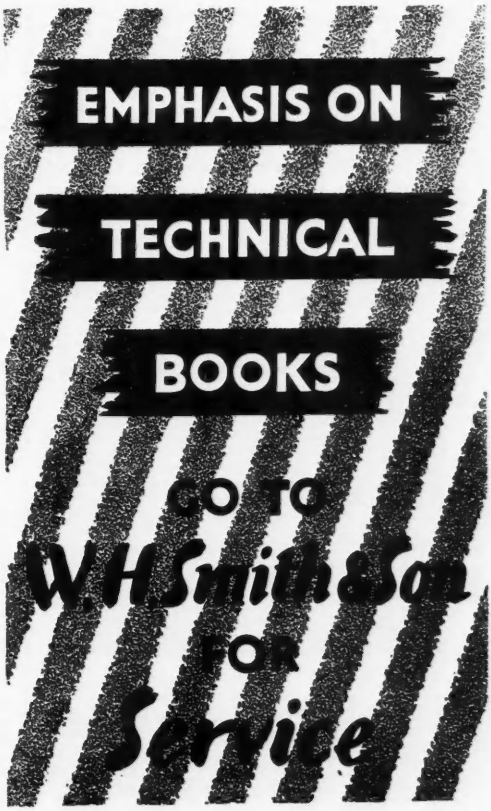
Contributing to these three ordnance programs, the work of the Naval Ordnance Test Station is concentrated in four primary development fields, rockets, guided missiles, aviation ordnance and underwater ordnance. In support of the development programs, the Naval Ordnance Test Station also maintains a vigorous applied research program and sponsors and conducts basic research from which future development programs may be engendered. The Michelson Laboratory is the principal facility for this research work.

Emphasizing the objective of blending service, scientific and engineering personnel into one operating staff, it is particularly appropriate that the name of Michelson was chosen for this Laboratory. The late Dr. Albert Abraham Michelson, America's first Nobel Laureate in physics, had himself what appears an ideal blending of service and scientific perspective. Dr. Robert A. Millikan, Professor of Physics Emeritus at the California Institute of Technology, in his memorial address in honour of Dr. Michelson stated in part, " Michelson, pure experimentalist, designer of instruments, refiner of techniques, lives because in the field of optics he drove the refinement of measurement to its limits and by so doing showed a sceptical world what far reaching consequences can follow from that sort of process and what new vistas of knowledge can be opened up by it. It was a lesson the world had to learn. The results of learning it are reflected to-day in extraordinary recent discoveries in the field of electronics, of radioactivity, of vitamins, of hormones, of nuclear structure, etc. All these fields owe a large debt to Michelson, the pioneer in the art of measurement of extraordinarily minute quantities and effects."

The Michelson methodology,—the emphasis on refinement of techniques for exact experimentation,—is being exploited by the research workers of the Inyokern centre in an approach to the weapons program which has already resulted in the development of new instruments effective as primary tools of research. Typical are the ultra-speed photographic techniques developed by Dr. W. M. Cady, Dr. A. M. Zarem, Dr. Franklin Roach, and their co-workers, in the field of " Microtime Physics." With this equipment precise observations can be made of changes in physical systems which occur during intervals as short as one hundredth of one millionth second.

An interesting step has been taken by the U.S. Navy in its decision to make a research laboratory the nucleus of operations at a desert station whose primary objective is defined as a weapons development program. Principal consideration in making this decision was the idea that development engineering is inseparably dependent on the research laboratory. Most important, it is dependent on being a part of the research atmosphere which stimulates radical approach to new development horizons as distinguished from the approach whose objective is to effect improvement in old technique. The history of the last war is sufficient to establish the utter reliance of any defense program which is to be effective in the future on its capacity to exploit promptly new concepts and new results of science. The best chance for accomplishing the purpose should come from close association of the research and development groups. Although the laboratory for research is a necessary part of a well-equipped development centre, it is obvious that no single laboratory can do more than a small part of the foundational research for any zone of development work. The laboratory is essential therefore for the additional reason that it offers qualified guidance and immediate contacts with the whole world of foundational research.

It is a part of the job of a weapons centre to contribute to the national effort to maintain clear technical perspectives and well co-ordinated work programs. In a very broad sense, the success of the entire national program for preparedness, based on the doctrine of peace by strength,—involves problems of over-all planning and co-ordination which are extremely complex. It is not clear now, for example, what allocations of our national effort to the various factors contributing to national strength will produce best results. Finding the answer to this question certainly requires study which goes well beyond the determination of desirable stockpiles of weapons and it must go beyond the matter of determining best facilities for developing new weapons. It must be concerned with industrial progress, with progress in the world of human relations and with training men for leadership in the many zones in which a strong nation must be strong. Too much effort in any one of these directions at the expense of the others may produce less than optimum results.



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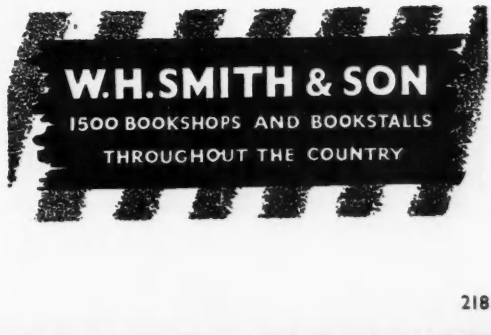

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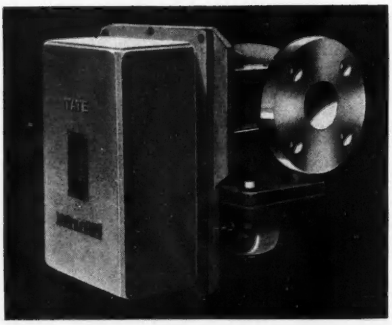

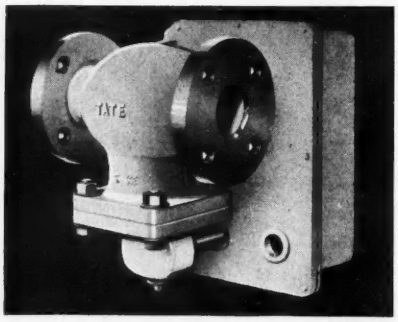

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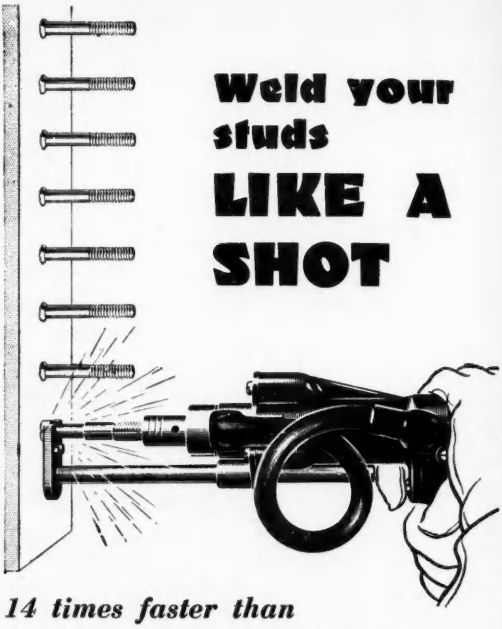

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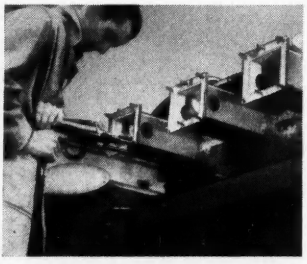

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As a part of the facilities for preparedness work of the most general sort,—along with the industrial laboratories, the factories, the universities, and other training, research and development centres,—the Michelson Laboratory has an important place. It is its job, for example, not only to develop new components for weapons and to do applied research work on which those developments can be directly based, but also to share the responsibility for training young people in the various technical fields, and to collaborate in the conduct of basic research on which technical process and much of the economic progress must eventually depend. There is now no doubt that a contribution to the scientific foundation for a weapons program is very likely to be a contribution to the scientific foundation for a whole zone of industrial development.

The measure of economic worth of the research laboratory (of a weapons development centre) can be positive in an absolute sense. It has a degree of responsibility in helping with the program to build, in proper balance, all components of national strength, including those which depend on the faith of people in government facilities skillfully operated. It can contribute to the solution of the planning problems inherent in dealing with the general problem of war and peace. These problems are in some respects not unlike those which face the planners of industrial enterprises, or the operations experts who prepare for combat, or the players of an elaborate game of some kind. Requirements for successful competition are remarkably similar in respect to their basic planning techniques. In each instance, there is sound reason for systematic study of the structure of the problem, the possible combat sequences, and the consequences of alternative choices in procedure. The available tools for such a study, besides operational experience, include the tools of engineering, mathematics, economics, biology and psychology. It is believed there is hope of gaining major advantages in a national preparedness program,—a program to avoid war,—by exploiting these techniques in building a strong nation.

It seems quite reasonable to expect that adequate progress toward the prevention of war through properly balanced strength will eventually be found also to have produced that kind of national strength necessary for highest levels of normal living in a world at peace.

British Aircraft Factories Working on more than 100 Different Types. A survey of the activities of the 30 principal airframe companies and seven aero-engine manufacturers in the United Kingdom discloses that there are more than 100 aeroplanes and at least 40 engines in production, under construction, in process of development or in an advanced stage of design—excluding an unspecified number of heavy jet bombers and new night fighters which are on order for the Royal Air Force, and certain aero-engines which are still on the secret list.

Of the sixty aeroplanes in production, nine are jet-propelled and three have turbo-prop engines. Of the 25 engines in production, five are turbo-jets and four turbo-props. Excluding aeroplanes which are serving as flying test beds for new engines, nine of the experimental aeroplanes now flying are jet-propelled and four have turbo-prop engines. Of 18 prototypes under construction, five are designed to take turbo-prop engines and five will have jets. Experimental engines now being flight-tested include two turbo-props and one turbo-jet; those under construction or on bench tests include five tubo-prop engines and one turbo-jet.

These figures supply ample proof that Great Britain is determined to make a thorough exploration of the field which can be covered by turbo-prop aero-engines. The range covers types for military and for civil use, and outputs varying from 500 to 5,000 shaft horse-power.

Gamma Rays for Industry. Eleven thousand copies of a paper dealing with radium and radon for industrial radiography, have been distributed by the Ministry of Supply recently.

Greater production of radium and radon at the Government-owned Radiochemical Centre, Amersham, will enable more extensive use of gamma radiography as an inspection aid to industry. It has been used for some time for examining steel castings and weldings and the positioning of inner components of complex mechanisms; there are, however, many possible applications.

Firms contemplating using gamma rays for the first time, should consult the National Physical Laboratory, Teddington, on precautions to be adopted in handling radioactive materials which do not involve much expense or difficulty.

Steps are being taken to encourage educational organisations to make provision for the demands of industry for training in gamma radiography.

New System of Speed Regulation. The General Electric Co., Ltd., is building for the Park Gate Iron & Steel Co., Ltd., Rotherham, two motor-driven turbo type blowers, each having an output of 25,000/30,000 cu. ft. per minute at 10/15 lbs. per sq. in., with grid controlled mercury arc rectifier equipments, a 3,900 kVA power transformer, high tension switchgear and cables.

Each blower is driven through speed increasing gears by a 2,200 h.p., 770/580 r.p.m. D.C. motor, which is supplied from a 1,740 kW, 750 volt, grid controlled rectifier equipped with additional anodes for supplying the motor fields. The rectifiers are connected direct to the motors without any intervening D.C. switchgear, protection against faulty operation being provided by field failure relays, anode fuses, and overload relays which trip the high tension A.C. circuit breakers.

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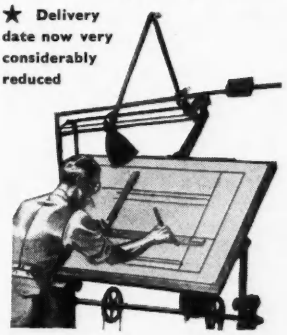

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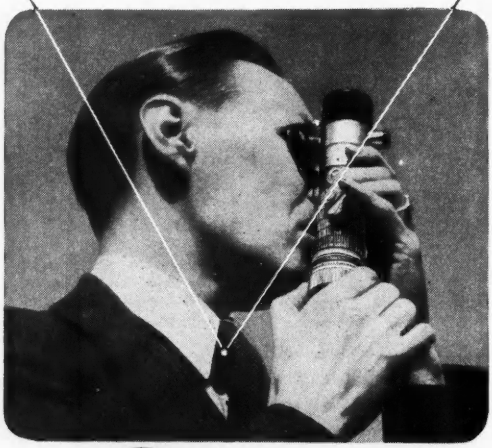

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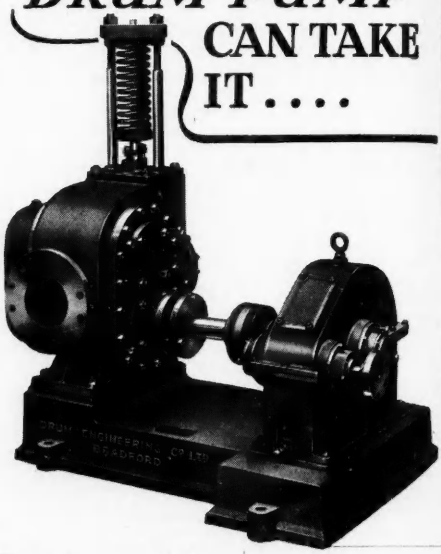

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thereby automatically maintains the volume of air at the present figure selected by the operator. It is believed that this is the first time grid controlled rectifiers in conjunction with "Reavell-Askania" regulators have been applied to blast furnace equipments in this country.

Alternative sources of A.C. supply to the rectifiers are also available, one from the B.E.A. 11 kV system and the other from the Steel works own 3.3 kV generating plant through a 3,900 kVA step up transformer. The high tension metal-clad switchgear is, therefore, of the duplicate busbar design, with 250 MVA spring-operated oil circuit breakers and oil immersed wing isolators for isolating either of the two supplies.

Standard Wrapped Bushes. The range of Standard Wrapped Bushes marketed by The Glacier Metal Co., Ltd., has been extended to include bushes with 3-in. bores. There are now 198 English sizes in the range, from ⅜-in. to 3-in. bores, and 127 Metric sizes, from 10 mm. to 58 mm. bores.

These bushes, made in bi-metal (steel-lined with lead-bronze)—a combination of high mechanical strength with an alloy possessing exceptional anti-frictional properties—can be used in practically unlimited number of applications.

Each size is made in large quantities by economical production methods, so costs are low, particularly in comparison with the normal solid type of bush. All the standard sizes are stock items and can be supplied immediately by the main London factory, in addition to branches at Manchester and Glasgow.

An up-to-date wall chart has been published recently and copies can be obtained on application to The Glacier Metal Co., Ltd., Alperton, Wembley, Middx.

ENGINEERING EXHIBITION—ZURICH

A "prestige" show of British engineering and allied industries has been planned to take place in the Kongresshaus, Zurich, from September 10-20, 1949.

One stand will be devoted to British gas turbine development on which more than a dozen of the firms concerned are combining. The remaining space in the exhibition will be available to manufacturers of engineering products, tools, gauges, measuring instruments.

BRITISH STANDARDS

(*Copies of British Standards may be obtained from the British Standards Institution, 24, Victoria Street, London, S.W.*1)

Analysed Samples for Metallurgical Analysis (B.S. 1548 : 1949). For some years analysed samples in the form of metals, alloys and oxides have been available for comparison purposes in connection with metallurgical analysis, and in recent years the demand for such samples has increased due to the industrial demand for greater analytical control.

This British Standard has been prepared to cover the general conditions to be complied with relating to the preparation, analysis, packing and storage of such samples in order that they may qualify for the use of the B.S.I. certification mark. For these purposes an analysed sample is taken as a homogeneous sample intended for comparison purposes in metallurgical analysis. PRICE 1/-.

Capacitors for Intrinsically-safe Circuits. (B.S. 1551 : 1949). In response to a request by the Ministry of Fuel and Power the British Standards Institution has just issued a British Standard for the construction of capacitors primarily intended for incorporation in intrinsically-safe circuits.

At the time that the Ministry of Fuel and Power's request was made, small capacitors were chiefly used as a means of spark suppression in remote-control circuits which were designed to be intrinsically-safe in accordance with the definition of that term given in B.S. 1259, Intrinsically-safe electrical apparatus and circuits.

As an outcome of recent research, however, alternative methods of spark suppression have been found for such circuits but there are other applications such as the coupling of an intrinsically-safe telephone circuit to the G.P.O. system. The capacitors for such a use are covered by this new standard, which deals with the dimensions of the containers and gives full details of tests. PRICE 2/-.

BOOKS

"Colour Conscription for Industry." Publishers: British Paints Ltd., Portland Road, Newcastle-on-Tyne, and Leadenhall Street, London, E.C.3. 22 pp. ring binder, fully illustrated in colour. PRICE: 10/6.

In the past few years, the far-reaching influence of the proper planning of colour in industry has been recognized as contributing directly to workers' efficiency, safety, comfort and morale. It has been realized also that factory lighting, to be fully effective, should be considered in connection with colour in factories and on machines. To cater for the increasing demand on the part of industry for specialists in this type of work, the Council of Industrial Design, jointly with the British Colour Council, organized a lecture and study course on the subject last November.

Industrial designers, architects, industrial psychologists, welfare officers and all others professionally interested in stimulating production, would greatly benefit by reading and putting into practice the recommendations contained in the booklet under review.

The rewards may be seen in increased production, more accurate work, less fatigue, reduced spoilage and improved morale.

This booklet, written in non-technical language and excellently illustrated, deserves to be widely read. Complimentary copies are available for principals from the publishers.

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The Rate for all classified advertisements is 6d. per word ; in bold print 9d. per word ; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

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